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VOL. 28

1922

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J. R. McCOLL

President 1922

American Society of Heating and Ventilating Engineers

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VOLUME 28

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TRANSACTIONS
OF
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 613

THE TWENTY-EIGHTH ANNUAL MEETING
1922

FOR its Twenty-Eighth Annual Meeting the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS again convened in New York, although it established a new precedent by holding the sessions in a hotel instead of at the Engineering Societies Building as has been the custom in former years. This arrangement permits the combining of the social headquarters with the professional. The facilities offered by the Hotel Pennsylvania were entirely suitable and the experiment proved to be a great success.

The unusually large attendance of members from the Middle West and Canada testified to the spread of the Society's influence. Much interest was displayed by local Chapters and the desire to extend the activities of the Society through the organization of additional chapters was evident. Altogether there were 382 present, of whom 223 were members, 160, guests, and 66 women.

The entire schedule of the Meeting was arranged so as to most harmoniously adjust the professional and entertainment programs. Registration headquarters were opened Tuesday morning, January 24, in the foyer adjoining the southeast ballroom, where the sessions were held. The business meeting took place in the afternoon and—at the evening session, following a presentation of a paper on the Mechanical Equipment of Hotel Pennsylvania by the designer, Arthur K. Ohmes, a tour of inspection was made of the mechanical departments.

A program truly representative of the entire field of heating, ventilating, and air conditioning made the meeting an outstanding one in the history of the Society. The most widely attended single session was that devoted to Research, showing the great interest which the entire membership feels in the Laboratory at Pittsburgh.

PROGRAM

FIRST SESSION

Tuesday, January 24, 2 P. M.

Business Session:

Announcement of Quorum
Appointment of Tellers of Annual Election
Address of President
Report of Council
Report of Secretary
Report of Treasurer
Reports of Committees:
- Committee on Chapters
- Committee on Legislation
- Committee on Code of Ethics
- Committee on Memorial to J. R. Allen
- Commitee on Revision of the Constitution
- Committee on Furnace Heating
- Committee on Steam and Return Main Sizes

Unfinished Business
Report of Tellers of Annual Election
New Business

SECOND SESSION

Tuesday, January 24, 7:30 P. M.

General Session:

Paper:
Heating, Ventilating and Mechanical Equipment of the Hotel Pennsylvania in New York City, by Arthur K. Ohmes.

Paper:
Improvements in the Process for Cleaning and Drying Air Mechanically, by Wm. J. Baldwin.

Topical Discussion:
Effect of Wind on Heating of Buildings (at request of Standardization Committee of Heating and Piping Contractors National Association).

THIRD SESSION

Wednesday, January 25, 10 A. M.

Ventilation Session:

Paper:
Standardized Method of Measuring Fan Delivery, by E. N. Fales.

Paper:
The Importance of Pure Water Supply in Humidifying Systems, by Fred F. Bahnson and Leona Maloney.

Paper:
Ozone in the St. Louis Schools, by E. S. Hallett.

Paper:
Hospital Ship Ventilation, by R. C. Holcomb.

FOURTH SESSION

Wednesday, January 25, 2 P. M.

Research Session:

Report of Committee on Research.

Paper:
Current Research in Heating and Ventilating, by F. Paul Anderson.

Paper:
A Study in Heat Transmission with Special Reference to Building Materials, by F. C. Houghten.

Paper:
Temperature, Humidity and Air Motion Effects in Ventilation, by O. W. Armspach and Margaret Ingels.

Paper:
Oxides of Nitrogen Produced by Ozone Apparatus, by G. W. Jones, W. P. Yant and O. W. Armspach.

FIFTH SESSION

Thursday, January 26, 10 A. M.

Industrial Session:

Paper:
The Control of Blower Motors, by Henry H. Issertell.

Paper:
Dehydration and Freshening of Codfish, by Henry W. Banks, 3d.

Paper:
The Under-Feed Stoker, by Frank A. DeBoos.

SIXTH SESSION

Thursday, January 26, 2 P. M.

Heating Session:

Paper:
Heating the Average Home Economically, by P. J. Dougherty.

Paper:
Heat Transfer for Conduction and Convection, by W. K. Lewis, W. H. McAdams and T. H. Frost.

Paper:
Tests of Humidity Conditions in a Residence Heated by a Warm Air-Furnace Using Recirculated Air, by A. P. Kratz.

REPORT OF COMMITTEE ON CODE OF ETHICS

TO THE MEMBERS OF THE SOCIETY:

The proposed Code of Ethics governing the engineering profession which has been under consideration for some time by the Committee representing the five national engineering societies, namely *The American Society of Mechanical Engineers; American Society of Civil Engineers; American Institute of Electrical Engineers; American Institute of Mining Engineers* and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, has finally been completed by the Joint Committee and is submitted herewith as a part of the Committee's report.

The work of the Committee was particularly difficult because it was felt that a Code of Ethics that would meet with universal approval would necessarily be a very short code, expressing in as few paragraphs as possible the essential matters which the members of the Committee felt should be included in such a Code of Ethics, and that having definitely decided on this Code of Ethics, that the interpretation and introduction of the Code into practical service among the various societies should be taken up by Committees of each Society or association and the Committee

should then supplement the Code itself with their interpretations of the various articles and their explanations as to why they were adopted and incorporated in the Code.

It is believed that all members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS who study this Code carefully will agree that the work of the Joint Committee has been well done.

It now remains for reasonable and fair interpretations of the Code to be put into effect by committees to be appointed by the various societies represented in the creation of the Code itself, which marks a great step forward in the placing of the Engineering Profession on a higher plane and should be the means of creating a much more favorable opinion of the Engineering Profession broadly by the public than has heretofore existed.

FRANK T. CHAPMAN, *Chairman,*
STEWART A. JELLETT,
PERRY WEST.

REPORT OF THE JOINT COMMITTEE ON A CODE OF ETHICS FOR ENGINEERS

The Joint Committee consisting of representatives of the *American Society of Civil Engineers,* the *American Institute of Mining Engineers, The American Society of Mechanical Engineers,* the *American Institute of Electrical Engineers,* the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, appointed to consider a Code of Ethics for engineers recommends, after deliberate consideration, that each participating Institute or Society adopt the short simple Code of Ethics following this report.

The Committee further recommends that the following method of interpreting and administering the Code be adopted by each participating Institute or Society after any necessary provisions have been made in the Constitution and By-Laws of the organization.

The President of each Society or Institute shall appoint a *Standing Committee on Professional Conduct,* to administer the Code of Ethics. The duties of such a committee shall be to interpret the Code and to render opinions on any cases of questionable conduct on the part of members that may be submitted to the Committee. These interpretations shall be reported to the Executive Board of the Institute or Society who may approve these interpretations, or take such other action as may seem just and necessary. The reports of the Committee on Professional Conduct when approved by the Executive Board, shall be printed in abstract *and in anonymous form* in the Institute's or Society's monthly journal for the instruction and guidance of fellow members.

This Committee on Professional Conduct shall be appointed in each Institute or Society by the President holding office at the time of the adoption of this Code and shall consist of five members, one appointed for five years, one for four years, a third for three years, a fourth for two years and a fifth member for one year only. Thereafter, the Presi-

dent then holding office shall appoint one member annually to serve for five years, and shall also fill any vacancies that may occur for the unexpired term of the member who has withdrawn. These appointments shall be made from among the older members of the Institute or Society, so that advantage may be taken of their mature experience and judgment. The Committee after appointment shall elect its own chairman and secretary. The Committee shall have power to secure evidence or other information in any particular case not only from the organization's own members, but if it should seem desirable, from men in other professions. The Committee may also appoint sub-committees to consider certain cases when deemed necessary.

This Committee shall investigate all complaints submitted to it by the Secretary of the Institute or Society bearing upon the professional conduct of any member and after the member involved has been given a fair opportunity to be heard, the Committee shall report its findings to the Executive Board of the Institute or Society. This report may in some cases suggest certain procedure of the Executive Board.

The Executive Board of the Institute or Society shall have power to act on the recommendation of the Committee on Professional Conduct, either (1) to censure by letter the conduct of the member who has acted contrary to the Code, if the breach is of a minor character, or (2) to cause the member's name to be stricken from the roll of the Institute or Society.

Copies of all reports made by a Committee on Professional Conduct to the Executive Board of each Institute or Society shall be furnished to each other Committee on Professional Conduct administering the Code. This will keep each Committee advised of the interpretations of other Committees, and in time an extended interpretation of the Code can be written based on the reports of the various Committees on Professional Conduct.

As interpretations of the various Committees on Professional Conduct administering this Code may vary at times, it is recommended that the chairmen of these Committees of the various Institutes or Societies be authorized to act as a Joint Committee to review such differing interpretations and to bring them into unity with one another.

Respectfully submitted,

Joint Committee on Code of Ethics

A. G. Christie, *Chairman*, A.S.M.E.

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A CODE OF ETHICS FOR ENGINEERS

Engineering work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The Engineering Profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1. The Engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.
2. He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.
3. He will advertise only in a dignified manner being careful to avoid misleading statements.
4. He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.
5. He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.
6. He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.
7. He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.
8. He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.
9. He will cooperate in upbuilding the Engineering Profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.
10. He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

No. 614

IMPROVEMENTS IN THE PROCESS FOR DRYING AND CLEANING AIR MECHANICALLY

By William J. Baldwin, New York, N. Y.
Honorary Member

AT the Annual Meeting of 1917, the writer outlined as an improvement in ventilating apparatus, a new design of fan known as an aero-hydro-fan-separator, which is intended for the separation of water, or watery matter, or moist dust, from air. For a complete description of the original design the reader is referred to Vol. 23 of the Transactions (1917), page 211.

Fig. 1 shows recent developments in the fan separation which may be termed the Baldwin method of washing, drying and conditioning air. This apparatus may be described as a tandem machine, as there are practically two or more separators on the same shaft, but located in a divided chamber.

The first part of the machine (A) is a simple fan, or any other means of moving air. It has for its object the cleansing of air mechanically by directing the air, or gas, laden with cinders or dust or even with another gas, against the inside of a permeable revolving cylinder or rotor (a).

The inner wall of the permeable rotor is placed at such an angle to the incoming air that the air strikes it at about 120 deg. The heavy particles, carried by the air or gas, (cinders or dust) are therefore projected against the inner wall of *the rotor*, and the fan is so designed that the dust leaving its inner edges or *tip* has what may be called an accelerated velocity which is again accelerated by the greater forward (rotary) velocity of the permeable wall of the rotor.

The air leaving the fan of course, can have no greater velocity than its tip intensity justifies, and this velocity can only be carried forward until the heavy particles held in suspension can come into contact with the inner walls of the rotor.

When the substances held in suspension in the air receive the second or re-accelerated motion, they hug the inner surface of the rotor, until a particle or particles find the first holes, when they are pushed through at the intensified velocity, which is much higher than the longitudinal air movement through the separator from inlet to outlet.

The air (or gas) does not follow the dust particles through the holes in the walls of the rotor, as the chamber outside the rotor is a *closed-chamber*, there being no outlet from it to the outer air.

Presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1922.

Outside of the permeable rotor (a) there is a permeable screen (b) almost similar to the rotor (a), but stationary. Through this screen, the particles which pass through the rotor (a) may be said to be rubbed by the rotary motion of the air (between the rotor and the screen) into the *quiet-space,* where they drop down and go out at the outlet (c) to a closed tank, or other suitably closed vessel.

The rotary motion of the air in the space (d) between the rotor and the screen is not communicated to the air in the quiet-space (e). Heavy particles, like cinders, are thrown directly through the rotor and thence rubbed through the screen, where they run off into a box or tank; often as high as 94 to 96 per cent of the solid matter in the smoke, even with soft coal is thus removed.

When it is necessary to rid the air of infinitesimally fine substances, or of another gas, as condensable acid, greasy matter, or even an effluvium, a suitable fluid spray is used, which agglomerates the dust, etc., and mixes with or condenses the acids so that the air can be separated of any foreign substances, except perhaps the excess of aqueous vapor that may be in the air.

But even in the case of aqueous vapor, when accompanied by a dry dust, a great deal of the vapor disappears and goes with the dust or with the spray-water, or the mud-water, produced by the agglomerations.

When the air leaving the section (A) reaches the section (B) in a condition of saturation or semi-saturation, as when a hot spray is used (or when hot water under very high pressures or temperatures is used), the humidity of the air leaving the first section is generally too high to be returned to the building, and the object of the second part (B), as shown, is to free the saturated air of the moisture due to the action of the spray.

The second part of the machine (B) is composed of a cylindrical rotor, somewhat similar to the first, and attached to the central drum or shaft in any convenient manner. This second rotor or cylinder (f) is composed of suitable, hygrometric material, which is porous and for some purposes is composed of limy substances held on the inner surface of a metal, permeable rotor. The hygrometric material when very friable, may be put between two concentric permeable metal cylinders, or even an unglazed terra cotta cylinder can be used.

The air with its moisture is directed against the inner surface of the revolving hygrometric tube, whose substance seizes on and absorbs the moisture at its inner side, while the centrifugal force and action carries the moisture through the wall of the porous cylinder. The moisture appears on the outer side of the wall of the cylinder like drops of perspiration, from which outer side the drops are thrown against the stationary metal screen, where they pass into the quiet-space and so down, escaping through the pipe (g) at the bottom of the machine. Almost chemically dry air can be delivered at the outlet end of the machine (h).

The machine can be driven in any way desirable and in any direction.

The twin apparatus shown in the illustration is tandem so-called, and operated and driven on the one shaft, but there is no objection to there being three or more stages in the machine, if the same is desirable. The

first stage removes the dust, such as bronze dust, where bronze is manufactured, doing it in the dry stage so as not to affect the color of the bronze; the *second* stage is a water stage, where a spray is used to agglomerate the very fine bronze which escapes the dry stage, and the *third* stage, the hygrometric stage, is to remove an excess of moisture, so that the air can be returned directly into the factory and the heat in the air saved. This is particularly necessary in factories where there are relatively few operatives, and where it is necessary to move large quantities of air, and to return the air to the factory, particularly in the winter time, for the sake of the heat it contains.

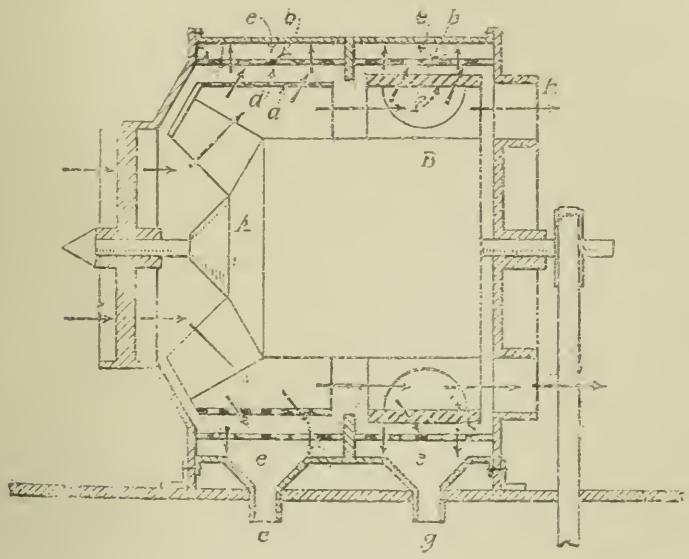

FIG. 1. THE BALDWIN FAN SEPARATOR FOR WASHING, DRYING AND CONDITIONING AIR

If 16,000 cu. ft. of air per min. is passed into a building and warmed to about 80 deg. fahr. from an average winter temperature, it calls for nearly 40 hp. to maintain its heat, and if this air is thrown away and fresh cold air is taken in again each time there is a loss of 40 hp. or thereabouts, continuously. It is obvious that if the excess air can be *returned* for the sake of the heat it contains, or for its excessive dryness, the expenditure of power is only that which will be necessary to operate the apparatus and move the air.

It is not factories alone, however, that the writer has in mind. It is any large space, with comparatively few operatives, and the process may be carried a great deal further than outlined, as in the case of the air in the holds of ships in the fruit trade, or in refrigerator ships and in storage warehouses and vaults the air can be *conditioned,* both in temperature and moisture, by this process and returned again and again.

Sugar-dust can be taken from the air and saved dry, to a very large

percentage (90 per cent), and what cannot be saved dry can be saved by being sprayed with a sugar-water or a syrup and afterwards evaporated and the avoidance of the evaporation of the enormous quantity of water necessary in the present process. The escaping dry air can be recirculated, thus saving a large percentage of the heat.

The apparatus requires very little room and will remain clean indefinitely.

No. 615

HEATING, VENTILATING AND MECHANICAL EQUIPMENT OF THE HOTEL PENNSYLVANIA IN NEW YORK CITY

By Arthur K. Ohmes, New York, N. Y.
Member

THE mechanical equipment of a building of the size of the Hotel Pennsylvania is on such a scale as to be almost incomprehensible. The magnitude of the apparatus and the amount of equipment required greatly exceed the figures that are familiar in ordinary building construction work. In view of the unusual size of the hotel it is believed that the following particulars may be of interest.

The Hotel Pennsylvania is the largest hotel in the world. It has 2200 rooms, occupying an area of about two acres, with cubical contents of 18,000,000 cu. ft., and a height of 27 stories, three of them below the street. Among its record breaking figures are the following: a dining room 142 ft. x 58 ft.; a plumbing pipe system of 111 miles; 26 elevators; ice and refrigeration plant of 195 tons; daily use in cooking of more than 50,000 cu. ft. of gas; laundry, washing and ironing daily more than 20 tons of table and bed linen alone; the world's largest private telephone exchange, with about 3000 stations in the house and 70 operators to handle the 24 hour volume of calls, which totals in a month, besides the incoming, interphone and long-distance calls, more than 10,000 outgoing city calls.

The number of guests averages 2800 to 3000 a day, with 2250 employees to serve them. Over 50 separate departments, shops and stores are housed in the establishment, including such widely divergent organizations as a carpenter shop, hair-dressing parlor, Turkish bath, post office, printery, grocery store, ice-cream factory, upholstery shop, refrigerating plant, engineering room, hospital and flower garden. The centralized management is so perfect, however, that all these enterprises are in effect but single units of a great business, where the methods of a high-power factory and low-cost department store combine for maximum results.

HEATING

The hotel building is heated with a low-pressure-steam vacuum-return-line system, the return lines being kept under a vacuum between 5 and 10 in. The steam pumps are of ample capacity and the automatic return

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valves are of sufficient tightness to easily maintain the higher vacuum. Ordinarily, the steam is carried at atmospheric pressure, but it is raised slightly during the cold weather and it is less in warm weather. In other words, a vacuum is maintained even in the steam lines.

Of special interest is the arrangement of the radiators, piping, etc., of the guest rooms, as illustrated in Fig. 1. Note that the radiators are hung on brackets; that all pipe connections come through the wall instead of through the floor, making possible the uninterrupted laying of carpets and easy vacuum sweeping below them. All guest-room radiators have

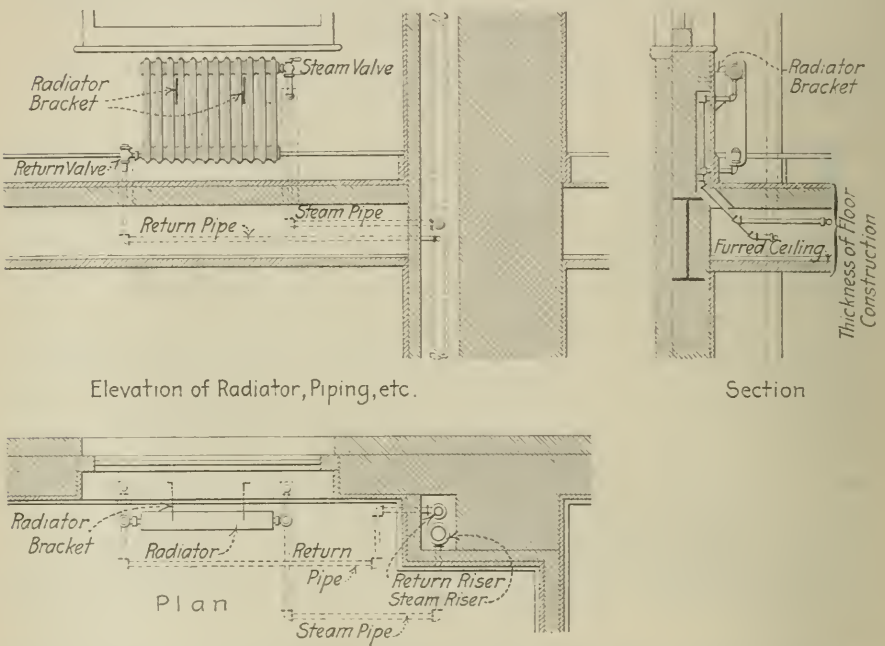

FIG 1. TYPICAL ARRANGEMENT OF RADIATORS IN GUEST ROOMS SHOWING PIPING CONNECTIONS.

been provided with fractional valves for hand control, but all public rooms have automatic temperature regulation.

All bathrooms are heated, whether inside or outside, and generally these radiators are arranged as shown in Fig. 3. The entire building has almost 5000 radiators, with a total heating surface of about 110,000 sq ft.

VENTILATION

The tabulation in Table 1 (page 20) gives some of the details of these tremendous and expensive, but so necessary equipments.

All fans are of the centrifugal multiblade type and are driven by direct-connected, direct-current electric motors. All air-supply systems, except for the engine room, have been provided with tempering and re-

heating stacks, there being provided about 16,000 sq. ft. heating surface for this purpose.

The air is taken from about 80 ft. above the street level where it is comparatively free from dust and consequently only four of the air-supply systems are provided with air washers. The air washers are run only in the summer months, because on dry days a considerable temperature reduction is secured by their operation. The construction

FIG. 2. THE HOTEL PENNSYLVANIA, NEW YORK CITY.

of the air washer is new as far as spray nozzles are concerned. There are only a few spray nozzles in each washer, which are supplied by small hydraulic motors. The water for these hydraulic motors is secured by a small electrically-driven pump.

All ventilating systems are selected so as to conform to the requirements of the temperature of the air supply as well as to the hours of working. The duct work on some of the more important exhausters is inter-connected so that repairs can be made in the original important units by having one exhauster take the air from two systems. There are literally thousands of air inlets and outlets provided, most of which are connected in the usual way by galvanized-iron ducts and flues, and it

would lead to a very long description if all of these systems were taken up in detail.

Therefore mention will be made of only the ventilation of the interior bathrooms as shown in Fig. 4. The vertical shafts act not only as shafts

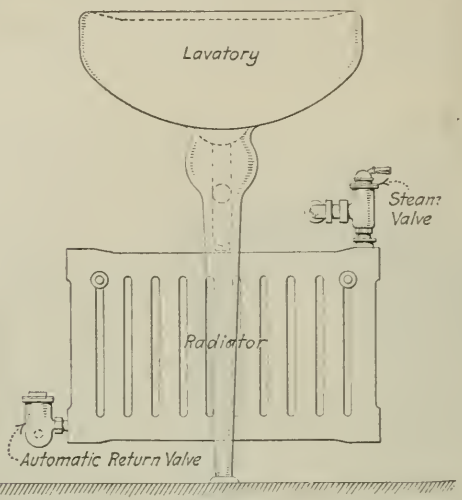

FIG. 3. ARRANGEMENT OF RADIATORS IN INTERIOR BATHROOMS.

for the pipe sysetms, but also as the exhaust shafts for the bathrooms themselves. In the top story, these shafts are connected by galvanized-iron ducts to the respective exhausters. The arrangement of the piping in the shaft is of special importance as well as the proper connection and size of the individual registers entering the shafts, and unless extreme care is taken an uneven air distribution will result. It is also necessary that the exhaust fans on these systems operate at all times, otherwise the circulation will be reversed in the upper bathrooms.

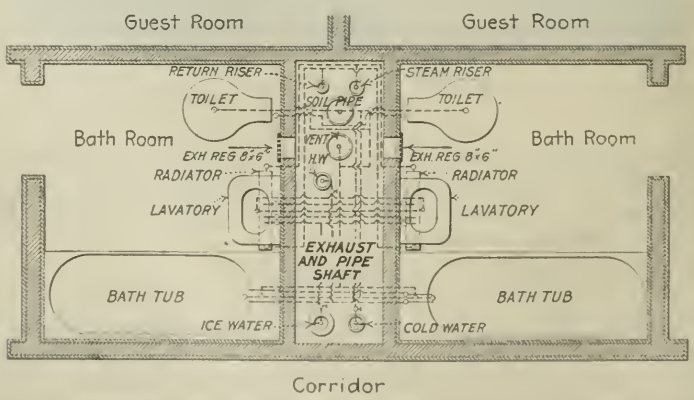

FIG. 4. COMBINED EXHAUST AND PIPE SHAFT SHOWING PIPING CONNECTIONS TO BATHROOMS.

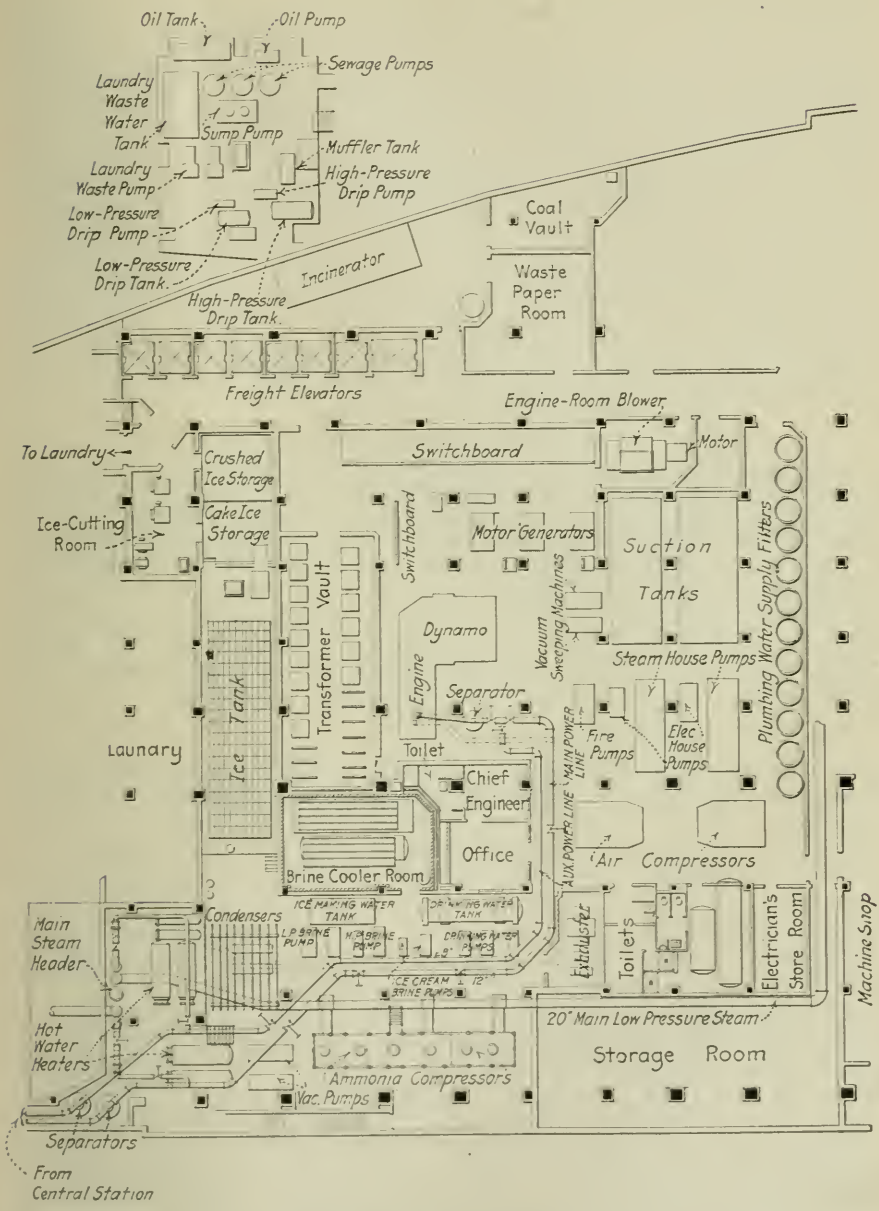

FIG. 5. PLAN OF SUB-BASEMENT SHOWING LOCATION OF MECHANICAL EQUIPMENT.

ELECTRIC SUPPLY

The building is supplied with alternating current at 11,000 volts from a power station of the Pennsylvania Railroad in Long Island City, which is about 4 miles distant. For power purposes, the high-tension current is reduced in transformers to 170 volts, which is then used for driving three 500-kw. motor generators to give direct current at 240 volts, this current being used for elevators, pumps, ventilating motors and the like.

For the illumination of the building alternating current is used, which is secured from another set of transformers, reducing the alternating current to 240 volts. The lighting circuits are three-wire systems.

There is also provided a 500-kw. steam-driven dynamo, which can generate direct current for power purposes in the winter months. In an emergency, this current can also be used for lighting in case the ordinary current supply should be interrupted, because emergency lighting circuits are provided with double-throw switches.

MECHANICAL EQUIPMENT

Refrigerating Plant.—There are provided three 65-ton ammonia compressors, of which two are in use 24 hours per day during the entire year. These ammonia compressors are driven by Corliss steam engines, so that it is easy to vary their speed to comply with the requirements in summer and winter, and day and night. The ammonia is condensed in double-pipe condensers. The cooling water used in the condensers is used for various purposes. The ammonia evaporates in double-pipe coolers of the three brine systems, as well as the drinking water cooler, the ice-making water tank and in the ice-making tank itself.

The brine systems are the following:

a. Low pressure for the three floors below ground and for the ground floor, operating at about 55 lb. pressure with a temperature of + 15 deg. fahr.

b. High pressure for the ball room, kitchen roof and kitchen floor pantries, operating at about 200 lb. pressure with a temperature of 0 deg. fahr.

c Ice-cream system for the manufacture and storage of ice cream throughout the lower floors, operating at about 56 lb. pressure with a temperature of – 7 deg. fahr.

Cooled drinking water is provided in all halls, kitchens, locker rooms, dining rooms and each guest bathroom; the operating pressure is about 160 lb. with a temperature of 40 deg. fahr.

The pipe systems of the low-pressure and ice-cream brine systems are designed as open-overflow systems, whereas the high-pressure brine and drinking-water systems are worked out as closed circulating systems. The frictional losses of all these pipe systems were carefully determined in order to secure perfect circulation and suitable pumping equipment.

The ice-making tank is capable of making per day about 100 blocks of ice of 300 lb. each. It is designed as a raw-water-ice system in which the ice is frozen under the injection of dehumidified and cooled air. An electrically-operated crane can lift two ice blocks at once out of the

ice tank when they are dipped into a hot-water tank to loosen them from the tanks and then hoisted out of the hot-water tank for dumping into the ice-storage room.

Plumbing.—The building has three water systems, one being of street pressure, which is mainly used for underground floors, another a low-pressure system for ground floor and up to and including the ninth guest-room floor, and a high-pressure system from the tenth guest-room floor up to and including the attic. In accordance with the water pressure maintained, there are provided hot-water heaters for the kitchen (street pressure) laundry (street pressure), lower guest rooms (low pressure), Turkish bath (low pressure), upper guest rooms (high pressure); the kitchen, laundry and Turkish baths have instantaneous hot-

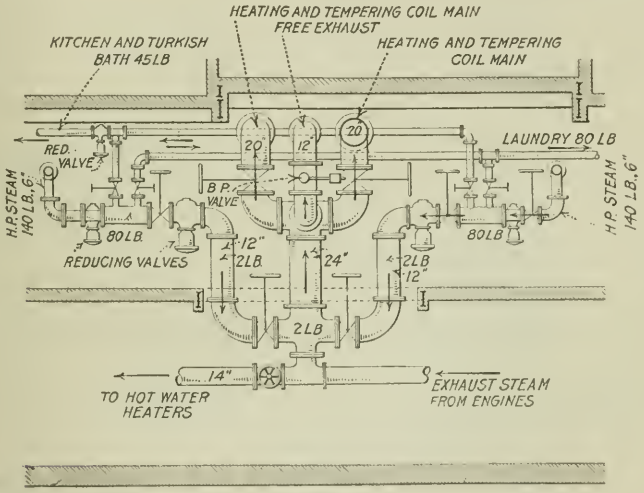

FIG. 6. ARRANGEMENT OF MAIN STEAM HEADER AND REDUCING VALVES FOR STEAM SUPPLY TO LAUNDRY, KITCHEN, ETC.

water heaters whereas the guest rooms have medium-sized storage heaters.

Compressed Air and Pneumatic Systems.—There is provided a very complete pneumatic equipment, which is supplied by two duplex steam air compressors driving air under 50 lb. pressure in a high-pressure air-storage tank. From this tank the air is taken for the raw-water-ice manufacture, temperature control and laundry. Small reducing valves maintain the pressure in individual systems as required. For the pneumatic system, the largest amount of air is required under a pressure of 5 lb. per sq. in. This compressed air is taken from the high-pressure air tank through a reducing valve into the low-pressure air tank. The bulk of this low-pressure air is required in sending messages, telegrams and letters from the main office on the ground floor to the floor clerks on the various floors; the mail is handled by floor clerks on each floor and is called for by the guests.

Elevators.—All of the elevators in the hotel are of the electric type.

There are provided in all 12 passenger elevators for the guests, six passenger elevators for the help, two freight elevators from the sub-basement up to the roof, four freight elevators from the sub-basement up to the first story and also two passenger elevators connecting the passage from the subway and the Pennsylvania R. R. Station below level with the first floor of the hotel.

Vacuum Cleaning.—The building is provided with a number of vertical risers so that each and every room can be cleaned with lengths of hose not exceeding 25 ft. The collecting pipes are located in the sub-basement where they are provided with frequent clean-out plugs. Tur-

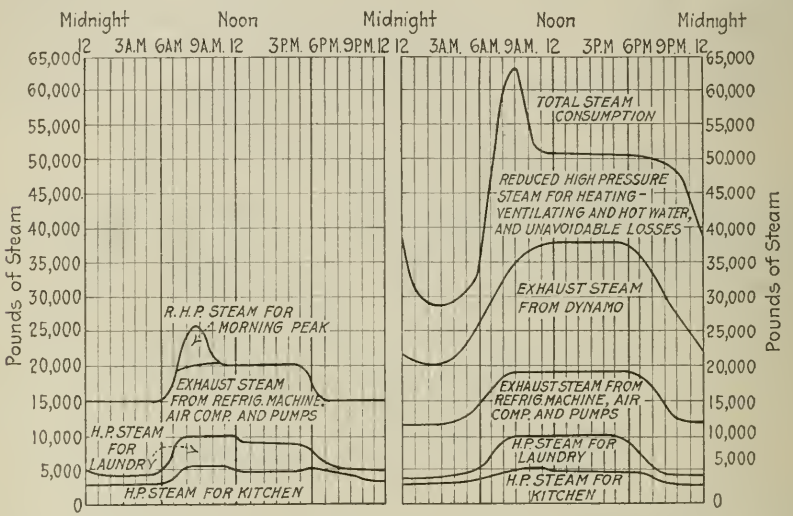

FIG. 7. CURVES SHOWING STEAM CONSUMPTION ON A TYPICAL SUMMER AND WINTER DAY.

bine blowers, having a capacity of about 2000 cu. ft. per minute, are provided for maintaining in the vacuum mains about 5 in. of vacuum, which is eventually reduced to 2 in. at the cleaning tubes. One machine is sufficient for the cleaning service. The cleaning hours are from 1 a.m. until 5 p.m. The public rooms are cleaned in the early morning hours, and in the day time the guest-room floors are cleaned.

Steam Power Machinery.—The above short description of the more important machinery equipment seems necessary in order to understand the following description of the supply of steam and electricity. The machinery equipment is mainly located in the sub-basement as shown in Fig. 5. The more important steam-power equipments are listed in Table 2.

STEAM SUPPLY

Steam under about 140 lb. pressure is obtained from the adjoining central heating station of the Pennsylvania R. R., which is located on 32nd St., between 7th and 8th Aves. This requires a length of steam

main of some 1600 ft. The main supply pipe is 12 in. diameter and the auxiliary supply pipe is 8 in. diameter. Both are worked out as a complete circuit in the hotel, as well as in the central station, and they are provided with a sufficient number of valves so that parts of this circuit can be shut off for repairs without completely interrupting the steam service required for heating, refrigerating, ammonia or air compressors.

The maximum steam consumption is about 90,000 lb. per hr. and under this condition there is fully 25 lb. loss of pressure in the auxiliary

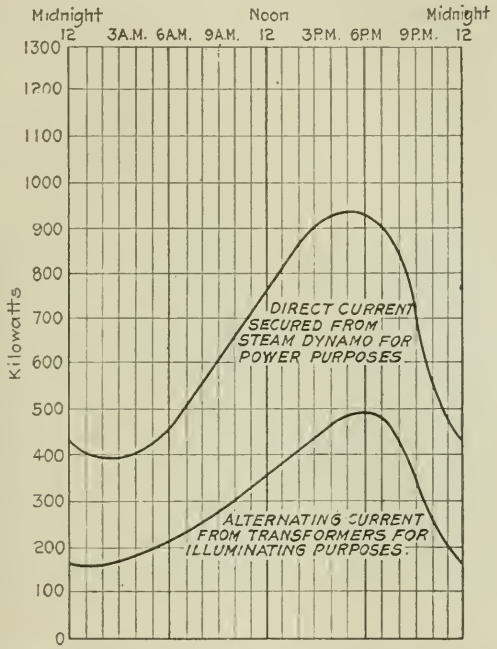

FIG. 8. CURVES SHOWING ELECTRIC LOAD ON A TYPICAL WINTER DAY.

main. Fig. 6 shows the main valve header as well as the arrangement of the reducing valve for steam supply to the laundry, kitchen and heating. All reducing valves are in duplicate, one side of the header connecting to the main steam-supply pipe whereas the other side connects the auxiliary steam-pipe line. The steam pressures which are generally maintained are indicated in Fig. 6.

STEAM AND ELECTRIC CONSUMPTION

A short statement of the idea prevailing when the installation was laid out, as well as a synopsis of how these ideas work in practice, may be of interest to the Society. As the matter stood at the time, it was possible to secure steam and electricity in unlimited quantities from the central station of the Pennsylvania R. R.; nevertheless it was desirable to save as much as possible on both supplies. Some calculations showed

that the low-pressure steam requirements for heating the hot water in the summer months were very nearly equal to the exhaust steam that would be secured from the ammonia and air compressors, and that in addition in early spring and late fall the exhaust steam from the house and brine pumps could also be used for heating the hot water; however, in the summer months the steam from these latter pumps could not always be used and it was consequently decided to install a steam pump for each of these services as well as an electric pump.

TABLE 1. CONDENSED SCHEDULE OF VENTILATING EQUIPMENT.

		Air Supply			Exhaust		
Denomination of Unit	Location of Rooms Ventilated	Cu. ft. per min.	Static pressure in inches for which unit was designed	Hp. of motor	Cu. ft. per min.	Static pressure in inches for which unit was designed	Hp. of motor
Engine room	Sub-basement	58,000	1	22	81,000	1	35
Laundry	Sub-basement	61,000	1	25	104,000	1	45
Service rooms	Sub-basement mezz.	58,000	1¼	28	54,000	1	30
Main kitchen	Basement	73,000	1¼	40	104,000	1	45
Grill room	Basement	26,000	1½	16	23,000	⅝	7
Lunch room	Basement	10,000	1¼	30	5,800	1	15
Cafe	Ground floor	.....	...	..	11,500	⅝	5.5
Main dining room	Ground floor	43,000	1½	33	54,000	1¼	35
Lobby, etc.	Ground floor	37,000	1¼	25	.....	....	..
Ball room	1st floor	52,000	1¼	30	47,000	⅝	26
North banquet room	1st floor	.....	...	..	11,500	⅝	5.5
South banquet room	1st floor	.....	...	..	11,500	⅝	5.5
Turkish bath	1st guest-room floor	20,000	1	8	17,300	¾	7
Guest bathrooms	1-15th guest room floor	.....	...	..	49,000	1	21
Guest bathrooms	1-15th guest-room floor	.....	...	..	70,000	1	35
Guest bathrooms	1-15th guest-room floor	.....	...	..	70,000	1	35
Guest bathrooms	1-15th guest-room floor	.....	...	..	70,000	1	35
Toilets	All floors	.....	...	..	87,000	1¼	45
Roof kitchen and ball room kitchen	1st floor and roof	.....	...	..	29,000	1	13.5
		438,000		257	899,600		446

Total 1,337,600 cu. ft. air and 703 hp.

TABLE 2. SCHEDULE OF MAIN STEAM-DRIVEN EQUIPMENT

Unit	Steam Cylinders			Equipment Driven
	Number	Diam. in inches	Stroke in inches	
Steam dynamo	1	24	40	500-kw. dynamo
Ammonia compressor	1	20	24	2 single-acting 14 x 21 in. ammonia cylinders
Ammonia compressor	1	20	24	2 single acting 14 x 21 in. ammonia cylinders
Ammonia compressor	1	20	24	2 single-acting 14 x 21 in. ammonia cylinders
Air compressor	2	12	14	2 double-acting 15 x 14 in. air cylinders
Air compressor	2	12	14	2 double-acting 15 x 14 in. air cylinders
House pump	2	16 & 24	18	2 double-acting 14 x 18 in. water cylinders
House pump	2	16 & 24	18	2 double-acting 14 x 18 in. water cylinders
Brine pump	2	14	12	2 double-acting 10 x 12 in. brine cylinders
Brine pump	2	9	10	2 double-acting 5 x 10 in. brine cylinders
Brine pump	2	6	6	2 double-acting 5½ x 6 in. brine cylinders
2 Vacuum heating pumps (each)	1	14	24	2 double-acting 24 x 24 in. condensation water cylinders
Fire pump	2	18	12	2 double-acting 10 x 12 in. water cylinders

Note that the smaller pumps and steam turbines for drip and drainage water, as well as laundry water are not included, inasmuch as they are small and are of no particular importance.

Further calculations showed that in the winter months, as soon as the heating system was in use, the exhaust steam from a steam engine could always be used if generating between 400 and 800 kw. It was, therefore, decided to provide a steam dynamo, which to a certain extent assumed the function of a reducing valve, but at the same time would generate electricity. Financial considerations and the desire to have the installation as simple as possible led to the decision of providing only one steam dynamo. This steam dynamo generates direct current only and is used for power purposes. It can also be used for emergency lighting purposes in case the alternating-current service for illuminating purposes should in any way be interrupted. A typical electric consumption diagram is indicated in Fig. 8. The steam dynamo is frequently in use in the month of May and again in September, in case on cold days steam heating is required. During one season of 200 days, this dynamo supplied on an average 7000 kw.-hr., which for a 24 hour period means an average load of 292 kw. When it is considered that unavoidable losses are very small with such a high average load, it is readily understood that the direct current from this dynamo is secured at extremely low cost.

It is not permissible to give the exact cost of steam and electricity which maintain in this particular installation nor is it of any particular importance. It is, however, permissible to draw some comparisons as to what economy would be the result if steam and electricity had been furnished by public service corporations in New York City. The following simple calculation would prevail: The dynamo requires about 50 lb. of steam per kw.-hr. All losses, including steam condensation and the loss of heat and steam through power generation, are well covered by 12 per cent of the total steam taken. Consequently, there is a charge against current generation of about 12 lb. steam per kw.-hr. generated. Since the steam would cost about $1 per 1000 lb., the steam charge would be for unavoidable losses 1.2 cts. per kw.-hr. Depreciation, maintenance and interest charges are approximately 0.4 cts. per kw.-hr. Oil and slight additional labor required will bring the cost of the kw.-hr. to not more than 1.7 cts.

The charge for a similar electric service for a kw.-hr., considering the large use of the building, would have been in the neighborhood of 3.1 cts. The saving, correspondingly, would be 1.4 cts. per kw.-hr.

Since there were generated by this machine during the winter of 1920-1921, a total of 1,400,000 kw.-hr., the saving in electric current bill would be $19,600.

Fig. 7 indicates the proportionate amount of high-pressure steam used on a typical winter day and the high-pressure steam used on a typical summer day.

No. 616

STANDARDIZED METHOD OF MEASURING FAN DELIVERY

By E. N. Fales[1], Dayton, Ohio
Non-Member

In comparing various cooling fans as regards their aerodynamic characteristics, there appears to be no accepted standard of measuring the output. The writer, having occasion to secure such measurements, made use of methods familiar in wind tunnel and general aeronautical research, wherein, by giving consideration to the peculiarities of the air-flow produced, results of commendable precision were obtained. It is the object of this paper to set forth the features of the method as applicable not only to cooling fans, but also to the larger type propeller fans where the flow is not enclosed.

Summary of Recommendations.—Analyzing the spiral and accelerated flow of the blast, we find its cross-section a minimum, 0.75 diam. and its velocity maximum at about 0.5 diam. downstream of the blades. Here is the proper place to make our measurements. A suitable instrument for the purpose consists of a miniature pitot-venturi tube, such as is used on aircraft, giving a pressure reading one and one-half times the pitot tube alone. The manometer for reading these pressures, of the Krell type, is one familiar in wind-tunnel work, but it carries a special graduation enabling the observer to read off the velocity direct in feet per minute; so that the high precision of the aeronautical laboratory methods does not interfere with quick and easy reading.

ANALYSIS OF AIRFLOW

In order to properly discuss measurement of air flow, especially at low velocities, it is desirable to examine the characteristics of the flow. Beginning with the open fan, of which the cooling fan is typical, we can apply bodily to it the method of analysis used in aircraft propellers, for it is the case of an airplane propeller *tuning up* on the ground. We shall see further that the air flow from an open fan has dynamic characteristics quite analogous to an enclosed fan, and that the method of analysis is similar.

We find the flow to be much more complex than is popularly supposed (see Fig. 1 and 2). The fan takes in air axially from the rear, radially from the sides, and even, slightly, from the front at the tips. Acceleration

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is very rapid, from practically zero a few inches away, to maximum at about 0.5 diam. downstream of the blades; the inflow acceleration is about one-third that of the outflow. The boundary of the flow contracts in proportion as acceleration exists; until at the minimum section acceleration is complete, and atmospheric pressure obtains in the blast. Thenceforth it decelerates, and spreads out at atmospheric pressure.

Emphasis must be laid on the fact that the blast converges as it leaves the blades, contrary to the popular conception (Fig. 2). Beyond the minimum section it does diverge, but very slightly. The erroneous idea that the blast is divergent as it leaves the blades is attributable to the large volume of entrained air which surrounds the main blast. Of this there is a greater and greater amount downstream, acquiring from the blast such energy as does not go into heat by internal friction. Aside from the small secondary flow of entrained air, the outflow blast (near the fan) is as definite and clearly defined as a jet from a garden hose.

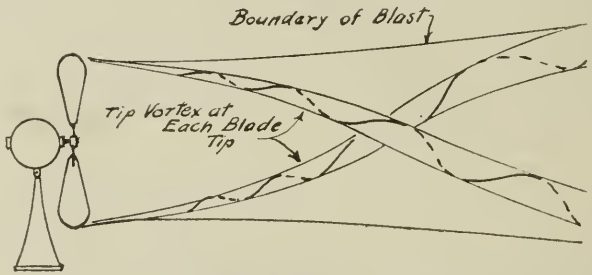

FIG. 1. SKETCH ILLUSTRATING FLOW THROUGH COOLING FAN, SHOWING SPIRAL FLOW AND TIP VORTICES

The blast is a spiral, as shown in Fig. 1. This is shown easily in an electric fan by fastening light threads to opposite radii of the fan guard, when they will be seen to be not parallel. The dynamics of the blast is, of course, that negative pressure is developed near the interior sufficient to counteract the centrifugal acceleration due to race rotation. The blast, furthermore, is enclosed in a dynamic *shell* which is somewhat analogous to a real duct. This dynamic shell consists of vortices trailing from each blade tip spirally around the blast. Fig. 3 shows such a tip vortex, wherein the blade is held stationary in a wind tunnel. (The visualization of the air flow is by the method of Caldwell, Grimes and the writer.)

The tip vortex, at the constricted portion of the blast, has an equivalent diameter which is of the order of 0.04 the fan diameter. Its structure is similar to that of a meteorological tornado, the center being at a lower pressure than the periphery. Its form, furthermore, is conical, and its rotation in revolutions per minute is high. It surrounds the fan blast spirally like the thread of a very high-pitched screw; meanwhile, its axis moves around the blast in step with the rotating blade.

The complex structure of a fan blast becomes evident. We see a spiral flow, which exhibits no centrifugal motion though it has rotation. This flow is surrounded by vortices, themselves of spiral and rotary structure. Nor is the *pitch* of these vortices necessarily the same as the pitch of

those air particles which partake of the *race rotation* of the main blast—it is rather on the order of double this latter pitch. Furthermore—due to viscosity—the outlying air, otherwise at rest, is entrained and takes up sympathetic motion which is of necessity spiral, though so feeble as to exhibit only its axial component.

The significant analogy, above mentioned, between this dynamic shell and an actual flue, has been corroborated by certain experiments. The actual pressure exchange familiar in a regenerative cone or *blower-chimney* has been observed even though the cone has been removed. That is, the phenomenon of a conical air-duct is reproduced without any cone being present, and can be satisfactorily explained if we assume the dynamic vortex shell to be analogous to a conical duct (see A. S. H. & V. E. JOURNAL, May, 1920, pages 409 and 410).

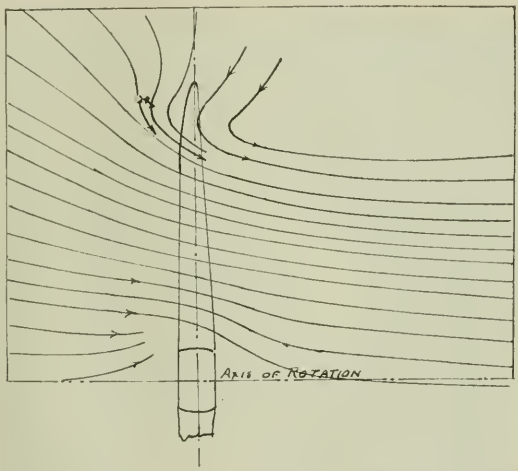

FIG. 2. FLOW LINES THROUGH PROPELLER FAN

We see, then, that a fan blast has a fairly definite boundary, even though unconfined by ducts. The problem of measurement at once becomes reasonable, provided we are able to take account of the spiral motion and provided the *dynamic shell* proves (as it does) to have comparatively small energy content.

VELOCITY MUST BE SPECIFIED AS MAXIMUM OR MEAN

From this preliminary analysis of the air flow, we can decide upon the most suitable method of measuring the delivery. We prefer to refer to *volume of delivery* rather than *velocity* in propeller fans, for the velocity varies at different radii and at different distances from the blades. It is obviously improper to rate a fan as giving a certain velocity unless we specify *maximum velocity* or *mean velocity at a given cross-sectional area* (see Fig. 4). Furthermore, in the specific case of cooling fans, it is volume of air rather than velocity which determines the drop of temperature of the cooling; for it can be shown that this drop is acquired by slow-speed air, and is not greatly improved by increased velocity.

In order to determine the volume, we are confronted with the need of making a traverse of the entire blast, point by point. To meet this need, at least in the case of small fans such as cooling fans, we require a minute instrument to give points of observation close together, say ½ in. More than that, we require an instrument whose manipulation is not laborious; one in which a precision of 1 to 5 per cent can be attained in commercial use, which is robust, and which reads directly in feet per minute rather than in *inches of water,* or *amperes,* etc.

USE OF AIRCRAFT VELOCITY INSTRUMENTS OF REDUCED SIZE

Having set these requirements, we may eliminate electrical and rotary instruments. The Thomas meter is for confined air flow; the hot-wire

FIG. 3. TIP VORTEX AS SHOWN IN WIND TUNNEL

anemometer is precise and compact, but is not very convenient in the hands of novices. The Caselli anemometer is bulky and unreliable, albeit a valuable *indicator* of flow due to its sensitivity. The pitot-static tube is more suitable. Complete satisfaction has been attained by the development of a venturi tube, in miniature of the full-flight tubes used by the Air Service, to be used in conjunction with a pitot tube, and a Krell inclined alcohol manometer. The one disadvantage of this layout is that the venturi tube has to be calibrated in a wind tunnel before use. Herein it is inferior to the pitot-static tube; which is easily reliable to a precision of ½ per cent without calibration when properly designed.

The advantage of the miniature venturi tube is the increased pressure recorded for a given velocity. While the pitot-static gives a pressure of $0.5\ p\ \frac{V^2}{g}$, the venturi gives $0.75\ p\ \frac{V^2}{g}$, or about 0.024 in water at 500 ft. per min. On a Krell manometer of 1 to 40 slope, using alcohol, this gives a reading of 1.4 in. It is seen that we are now in the realm of satisfactory precision. Actually it is the pulsation or gusts in the blast which limit the precision of reading.

Where the blast is large and a bulky measuring instrument is allowable, a larger pitot-venturi is to be recommended; for the *multiplying factor* increases with size. A pressure reading five times the pitot reading can be secured with an ordinary aircraft speed instrument, where

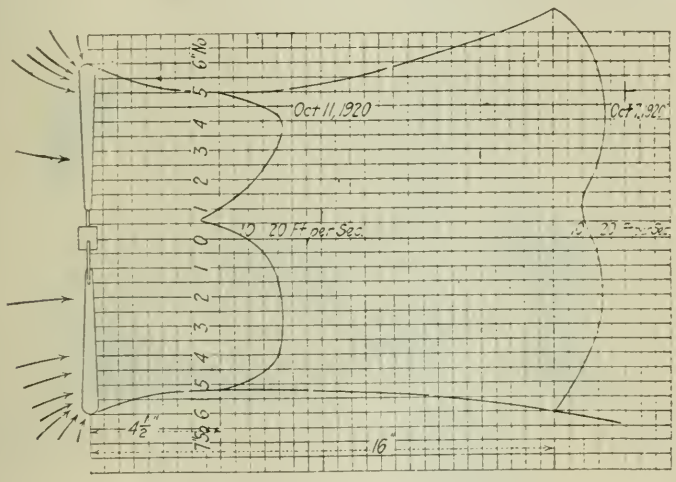

FIG. 4. DISTRIBUTION OF VELOCITY ACROSS BLAST OF COOLING FAN

the speed is fair. The venturi tube when immersed in air flow is an example of the *law of dynamic similarity*, wherein either size or velocity decrease results in performance decrease.

Fig. 5 shows a pitot-venturi tube and manometer set up for tests on the air delivery of a conventional fan. The photograph also shows, for comparison, a full-sized aircraft pitot-venturi tube. The operation is convenient and quick. The error due to slight angularity between pitot and venturi and pitch line is sufficiently small that its effect may be ignored in conventional fans.

The operator has only to move the pitot-venturi head to successive divisions as marked on the horizontal slide-scale; or on the vertical column, when the manometer reads directly in ft. per min., the corresponding velocity. Traverses of two diameters are covered thus, one vertical, one horizontal. The pitot-venturi stand can, if desired, be stood up at the 45 deg. angle, when two additional diameters are secured,—ample for commercial accuracy of volume measurement. No computations are necessary for the velocity measurement with this instrument,

and a complete analysis of air delivery may be effected, to an adequate commercial precision, in a small amount of time.

VELOCITY MEASUREMENT TO BE TAKEN AT HALF-DIAMETER DISTANCE FROM BLADES

Now as to the position in the blast where the measurements are taken. It is assumed that the air flow is unconfined. Methods have been used wherein the fan is placed in an orifice in the wall of a sealed chamber, an inlet or outlet being provided in addition to the fan orifice, the flow through this inlet or outlet being measured by suitable means. Such a method interferes with the air flow of cooling-type fans and produces erroneous results.

Measurements are not readily made at the inlet side of a propeller fan, for the inflow velocity is small. They are best made on the dis-

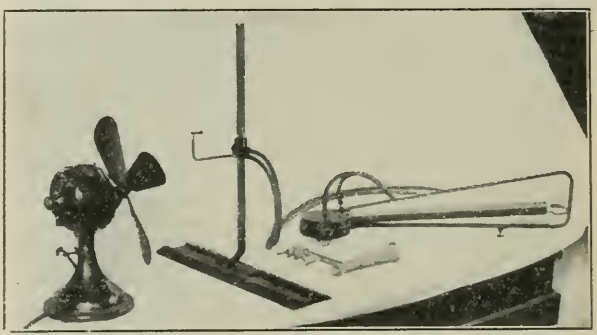

FIG. 5. METHOD OF TESTING FAN WITH SMALL VENTURI AND KRELL TYPE MANOMETER

charge side at the constricted portion of the blast about 0.5 diam. from the blades. It seems highly desirable, as a means toward standardization of measurement, that this portion only be used for the following reasons:

1. The boundary of the blast at this point is sharply defined, except where distorted due to the proximity of the base;
2. The velocity at this point is maximum, hence most easily measured;
3. The position is easily found, being approximately 0.5 diam. in front of the blades;
4. The static head is at atmospheric value.

EXHAUSTER FANS

Having analyzed the unconfined air flow of an open fan, we can follow through the same principles as applied to an exhauster fan. While this type has no air duct, being simply located in an opening in the wall, its pressure gradient differs from that of the open fan. In practice, such a fan is not used where pressure difference on either side of the fan is high; it is primarily used as a momentum fan.

The exhauster fan has but little inflow momentum and air in the

vicinity of the intake can follow streamlines of low radius, unlike that of the discharge side where the momentum is greater. Hence the ease with which air flowing in through windows can reverse its direction 180 deg. to pass out through the fan, leaving the main mass of air in the room unaffected. It is thus evident that an exhauster fan may prove inefficacious where open doors or windows permit the pressure difference to be lost. This is frequently noticeable in factories or toilets, where a window or other opening becomes a point of entry for return flow into the room; so that much or all of the delivery volume is air which has just come in at the window. Such a fan is only a device for producing

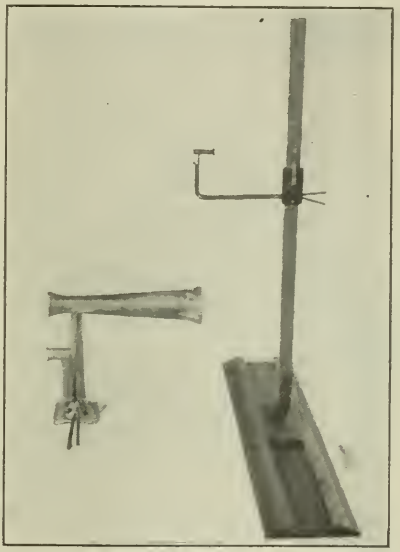

FIG. 6. VIEW SHOWING FULL SIZE AND MINIATURE PITOT-VENTURIS

a dynamic disc of pressure difference, which, when relieved by open doors and windows as above, is of small value. Many a wall ventilator can be found in summer, humming busily through the heat of the day, doors and window wide open. In this case its chief contribution to the general comfort is the psychological one to be had in its emission of a cooling noise.

Referring to the diagram of air flow in Fig. 2, the volume entering radially from either side is of major importance, as may be corroborated by blowing smoke into the tips of a cooling fan. A further and more striking corroboration is to bring a plane surface up at the rear, cutting off the axial inflow, when the delivery is seen to be only slightly impaired. Furthermore, such a pressure disk experiences a reverse flow near the tips, comprising air which is drawn towards the blade from in front, then turned 180 deg. on itself and propelled forward nearer the axis. Such reverse flow has often caused ventilating engineers anxiety when observed after installation of wall-ventilators. It is not, as is usually

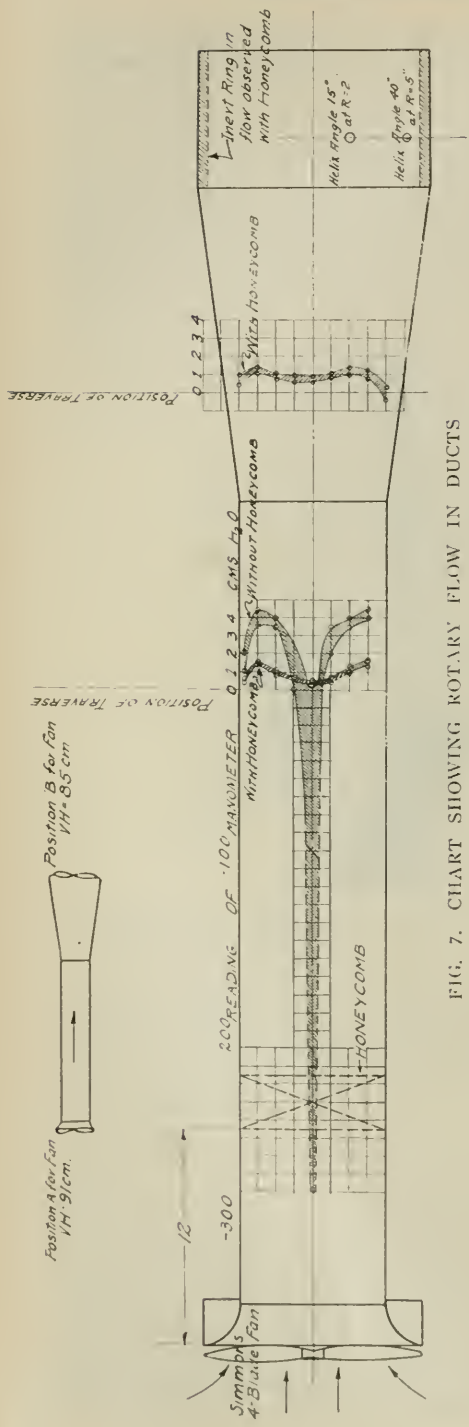
Inert Ring in flow observed with Honeycomb
With Angle 15° at R=2
Helix Angle 40° at R=.5
With Honeycomb
Without Honeycomb
With Honeycomb
Position of Traverse
0 1 2 3 4
Position of Traverse
0 1 2 3 4 cms H2O
Position B for fan VH = 8.5 cm
Position A for fan VH = 9 cm
200 Reading of 100 Manometer
Honeycomb
-300
12
Simms 4 Blade Fan

FIG. 7. CHART SHOWING ROTARY FLOW IN DUCTS

supposed, a matter of clearance between blade tips and orifice edge. All fans show this more or less, the ordinary cooling fan being a common example.

REVERSAL OF FLOW AT HUB

There is another cause of negative flow which deserves attention: It occurs in all types of fans to a certain degree, as well as in wall fans. This is the negative flow at center. The *dynamic pressure disc* above mentioned is not one of uniform unit pressure, but in most fans has its maximum intensity at two-thirds the radius where also the thrust and velocity are greatest. Even a carefully designed blade cannot maintain near the center a high pressure. In aircraft propellers the non-effective hub area may be 0.10 the disc area. Thus there is at best relatively small discharge at the hub. Frequently the pressure gradient is such that reverse flow actually occurs, a strong flow being visible toward the hub, near the axis. This air need not be assumed to pass backward through the blades; it is preferable to assume it receives centrifugal acceleration, and is subsequently discharged at a larger radius, moving thenceforth with the general air current.

USE OF BLANKED-OFF HUB

It is this consideration which leads to the use of a blanked-off hub; in the fan shown in the JOURNAL for May, 1920, page 411, the hub is blanked off for two-thirds the entire fan diameter. Furthermore, the blades are so shaped as to give a fairly uniform unit pressure. The head against which it operates is extremely high —16 in. of water. Thanks partly to the blanked-off hub, the fan is highly efficient.

The discharge blast of a wall fan contains a good deal of energy in it, which may be partially recovered, where highest efficiency is required. Thanks to the rotation of this blast, it is able in decelerating to follow beyond the constricted portion, a cone of high angle. Unlike a rectilinear flow, which demands a cone of 5 to 7 deg. for best efficiency, this flow may show favorable regenerative effect at cone angles more suitable for commercial use. By this means the air exhausted through an aperture of given size becomes equivalent to that of a somewhat larger aperture.

In order to measure the volume delivered from a wall fan, we must consider the features analyzed above. Since it is usually inconvenient to seal up the chamber of exhaustion, we may apply to the discharge blast the same method described for cooling fans; with the addition of a measurement of the static pressure step on either side of the fan. This is easily done by a delicate manometer.

PROPELLER FANS IN DUCTS

The flow of air in ducts, especially when the fan is of the propeller type and upstream of the measurement, must take into account, equally with the two types of fan previously mentioned, the character of the air flow. Fig. 7 is typical of the rotary flow thus produced. In securing a velocity traverse of the stream it will be found in some cases that the reading comes out negative at the center. This does not necessarily mean that there is a reverse flow; but it is attributed to depression due to spiral motion, and is quite analogous to the low pressure area so familiar in tornadoes. (In the latter instance the depression due to centrifugal acceleration is responsible for the lifting of roofs, etc.) In spiral flow emphasis must be laid on the non-uniformity of static pressure along a diameter. It is obviously improper to use openings in the side-walls without first calibrating.

The spiral angle depends on the fan efficiency, and on the pressure head against which the fan works. It may be sufficient to cause erroneous readings at the static head openings of the pitot-static tube, or the pitot-venturi tube (whichever is employed), if they are not oriented with the stream direction before reading. In aerodynamic work a swivelled head is sometimes used, which is free to take up, like a weather vane, the true streamline position. It is usual, however, to shift the direction of the instrument by hand, observing the manometer reading and thus finding the proper position. In some cases a silk thread fastened to the pressure instrument may be of assistance in finding the streamline. The axial velocity component for each reading is then the measured velocity divided by the sine of the pitch angle. We would prefer to measure the axial component direct, if some instrument were available for doing so. It is in such a flow that the hot-wire anemometer or the Thomas meter, when calibrated, is advantageous. Spiral flow of low pitch is to be avoided in ducts by inserting straightener vanes, whereby the wasteful internal friction is decreased.

Immediately down stream of the fan the air is accelerated and occupies a smaller cross-sectional area than further on. If the duct is not proportioned to correspond there results either a core or ring of air which does not partake of the general forward movement. This air takes

up backward motion toward the fan hub, or it may take up rotary motion: in any case it represents lost energy.

DISCUSSION

F. R. STILL. The proper method of testing ventilating or propeller fans, is a subject about which there has probably been more dispute over a longer period perhaps, than any other question about the use of fans or blowers. One maker follows a certain practice and every other maker has a different method. It is extremely desirable that they all adopt one method. There can only be one best way, so we want to establish what it is and get up a code to follow when testing such fans.

The National Association of Fan Manufacturers had a meeting in Buffalo some time ago, to which Dean Anderson, Mr. Lyle, Mr. McColl and myself were invited. We went there to obtain funds for the Research Laboratory, but we got into a fan testing discussion which resulted in a committee being appointed by the fan manufacturers to confer with a similar committee to be appointed by the Society.

The paper just read is a very valuable contribution and in some respects is quite extraordinary, but to my mind, from a casual reading of it, I do not think the method proposed is adapted to the requirements of the engineers of this Society. I base my conclusions somewhat on a personal experience I have had with some of the air testing engineers who have been sent up to our plant to conduct some experiments after they had left the aircraft service and had gone into practical lines of work, particularly in connection with air cooled automobiles. We several times have upset their previous conclusions completely. They have usually left us with the conviction that we were working along the right lines and so far as we know, they are still using the same methods as we have used for the past 25 years or more.

I think it would be inadvisable for the Society to do more than receive this paper, with thanks to the author for his valuable contribution, and leave the matter in the hands of the Committee on Standard Method for Testing Fans. As one member, I have a determination to see that the committee does some real work and I am hopeful inside of a year that we will have a code to present which the Society can adopt.

No. 617

OZONE IN THE ST. LOUIS SCHOOLS

By Edwin S. Hallett, St. Louis, Mo.
Member

At the 1920 Semi-Annual Meeting of the Society the writer presented a paper in which was described some of the results of the use of ozone in the ventilation of the public schools of the City of St. Louis. Investigations had been previously made by experiments on volunteers in the drafting room and in the homes of employees of the Board to determine the concentration necessary or desirable in ventilation. The chemical determinations of the output of the apparatus was made by Professor Anderegg of Purdue University, especially to check against the presence of substances other than ozone. It was especially desirable to be certain that there were no appreciable quantities of nitrogen compounds present. It seemed specially important to avoid the ionization of the nitrogen as the reports of some experimenters could not be reconciled upon any theory except that they had been dealing with a mixture of ozone and peroxide of nitrogen. While our concentrations were known in terms of grams per hour and parts of ozone in a million parts of air it was of little significance since there had been no standard of concentration established for ventilation. It was not and is not now desired to supply ozone in a concentration that may be called artificial. It is an uncontroverted fact that ozone exists in the air at sea level in unoccupied territory and in higher concentrations at higher altitudes. The fact that carcasses of cattle and horses dried up without decay on the western plains in the days of sparse settlement is explained only by the presence of the antiseptic qualities of ozone. It is well known that tubercular patients recovered rapidly in some of the southwestern towns in the early settlement days, whereas these places grown to cities are no longer resorts for this malady.

The early studies made preliminary to the introduction of ozone into the first school developed some interesting facts. Tests were made on a number of persons with ozone concentration many times that for ventilation. The effect was a roughness or slight irritation of the throat which disappeared in two or three days. The only external indication of irritation from very high concentration was a redness of the margin of the eyelids of a young man which cleared up entirely in a few days. No bad effects followed though the ozone was continued. Another case was a prominent baritone singer who had previously suffered some injury to his voice in a serious illness. His experience in the strong ozone in

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the drafting room at first was a perceptible huskiness but after a week he was sure his voice had recovered the original strength and purity of tone.

All the observations made seem to point to the fact that ozone is not an irritant to the mucous membrane in itself but the active oxidizing power of ozone attacks the secretions lying on the surface and imbedded within the open structure of the membrane and this byproduct produces some irritation and the forcible removal of the secretion may have a physical straining or rupturing effect on the tissues of the membrane. The reason for this belief lies in the fact that the irritations, where present at all, subsides within a few days with the high concentration still maintained. Another evidence is taken from the similarity of ozone to hydrogen peroxide, which attacks the secretions and extraneous matter but does not effect living tissues. It has the same irritating symptoms when applied to the delicate membranes of the throat but has no permanent effect except to cleanse the surface.

The important conclusion from the foregoing is that experiments to determine the effect on the human body in any concentration should be conducted for a long period, at least for several days, as nearly continuous as possible. Misleading reports have been made on the effect of ozone on bacteria. Bacteria vary in their tenacity of life about as the delicate orchid in comparison to the scrub oak. Undoubtedly ozone in the low concentrations for ventilation does destroy a large percentage of the germ life of the air.

The tests made in the Mullanphy School in St. Louis on April 14, 1921 with and without ozone showed less than half the colonies of bacteria with ozone as were found in the test without ozone. An average of 6 colonies per station were found with ozone on April 14th and an average of 7.7 colonies were found on April 19th, which would give better than 97 per cent rating on the *Synthetic Air Chart*. The fortunate circumstance is that all the disease-producing bacteria are easily destroyed and it is hoped that the half which were destroyed by ozone were those injurious to man. It is of no interest to the heating and ventilating engineer to know the concentration of ozone that would destroy all bacteria, if there were such a concentration.

We use no germicidal treatment of drinking water until B, coli makes its appearance, when in this country, liquid chlorine is used and we may be highly gratified that we have a convenient and harmless treatment of the air that destroys the bacteria that menace the health.

In 1919 the first ozone machine was installed in the air duct of the O'Fallon School. Many hundred visitors from all parts of the country have visited this school. The effects on the comfort and health of the teachers was so marked that in the summer of 1920 the ozone was made a part of the summer repairs in thirteen old schools and put in the specifications of three new ones about to be let. Nearly all these old schools were in the districts most needing it. In most instances the principal and teachers were not apprized of the change in the ventilation. Having studied the behavior of ozone in a frank and honest trial at the O'Fallon School it was desired to see how much weight the so-called *psychology* might have. The result was that in a few weeks most

of the principals were greatly pleased with the result. Many of the teachers volunteered the information that the improvement in the health of the children was marked. A principal stated that it was the first winter in many years that he had been free of colds.

The writer dropped in on the physical training supervisor at the Rose Fanning School during the first winter with ozone and asked her how she liked the new ventilation. She replied that she knew nothing of any change in the ventilation but as she pointed to her class of children seven to nine years old said: "You see what fine examples of health are here: a number of these were quite thin and pale at the opening of school in September. They have all gained in weight and the pale ones now have rosy cheeks as you see. I can not account for it. I have never seen anything like it before." In speaking to another teacher in the same school I asked her how much she had gained in weight during the term. She asked in considerable embarrassment why I wanted to know. I replied that the new system of ventilation generally resulted in increase of weight. She stated that she had gained 15 pounds but attributed it to her walking more. My rejoinder was that no doubt it did but the ozone gave her the invigorated feeling to want to walk after a day in school.

The experience with ozone at the Mullanphy School has attracted very wide attention largely because of the interest taken in it by Mr. Shackelford, the principal. Mr. Shackelford is a highly trained scientific teacher and lecturer. He lectures on these subjects in the Harris Teachers College in addition to his duties as principal of the Mullanphy School. In conjunction with Dr. William Weiss of the Hygiene Department they undertook to get the actual facts about the effect of ozone on the school. Records were kept of all absence on account of illness, the nature of the disease and the time lost from school on that account. As a check on this data the same records were kept of the absence at the Shaw School, the nearest school and in a similar district. The tabulation of these records is appended hereto.

At the February, 1921, meeting of the St. Louis Chapter the writer acting for the Commissioner of School Building tendered the Chapter the use of a school building for the purpose of making tests of heating and ventilation as an educational feature of the chapter meetings. This was immediately accepted and the acting director of the Research Laboratory of the Society, Dean L. A. Scipio was invited to be present and direct the tests. The date was set for April 14, 1921 and Secretary Obert and Mr. O. W. Armspach accompanied Dean Scipio in making the test. It was decided to make two tests on that day, that in the forenoon without ozone using the synthetic air chart as the basis. The afternoon test was made under same conditions except with the ozone on. The physicians of the Hygiene Department secured the agar dishes and exposed them and returned them to the City Bacteriologist for incubation and count. The CO_2 samples were taken and shipped to Pittsburgh for determination. The tests made on April 14, were with the fresh air intake open.

It was desired to have a test made with recirculated air and it was decided to make this on April 19th. This was conducted exactly as the previous one but with 65 per cent recirculated air, without the mist nozzles

in operation. The scrubber plates of the Warren Webster washer were wetted with special jets. A perusal of these charts reveals these important facts: The afternoon test with ozone gave one-half the average bacteria count as in the morning without it; that recirculated air gave 36 per cent less bacteria and 15 per cent less dust than all fresh air without ozone. Odors were pronounced to be entirely absent in the recirculated air test.

The teachers in this school are unanimous in stating that the new ventilation does relieve the fatigue of the day's work. An incident occurred a few weeks after this test that indicates the progress of ozone. The school holds an annual entertainment in which the parents association participates. The weather was warm and the throngs filled every seat and all standing room. The principal asked that the rule be suspended and the ventilating system be operated as he said the open windows did not relieve the stuffy condition even for rehearsals. As the crowds were gathering and the janitor was about to start his fans and ozone, some leading citizens came down and said: "We must open all windows tonight, it is hot." Mr. Louer, the janitor said, "We want to close the windows tonight and run the fans." "No," said the leading citizen. "That mechanical ventilation is all bosh. A big engineer told me there was nothing in it." "Suppose you go back to the Auditorium and let me operate for ten minutes and then if you want to open the windows I will shut down," said Mr. Louer. Mr. Leading Citizen went back and did not return to the engine room until the entertainment was over when he appeared and handed the janitor some bills and said, "It is worth this to me to know that there is such a system as this. It was delightful all through. I am coming over next week and take time to go through and see how this is done."

During the past summer ozone was installed in two old high schools. One of these, Central, is a very old structure, remodeled many times with the split system of heating. No school had more of the old school room and musty locker smell than this one. Mr. Douglass, the principal, is considered an authority on many subjects and advised extreme caution on ozone at the beginning. He was, however much pleased to have a try out in his old Central. After three months he says it is decidedly improved. The old odors are gone. The engineer who takes care of all the complaints says he has had nothing like half the complaints that he had received in former years.

There is no doubt whatever of the beneficial effect of ozone upon the health of the children as used in the St. Louis Schools as evidenced by the gain in weight and the very perceptible gain in ruddy color in the faces of all children and especially those that are pale and anaemic. The St Louis Board of Education maintains two open air schools with an attendance of 250 to 300 children. There are no tubercular children admitted but only such as have the anaemic symptoms which precede the open lesions of the disease. All the meals are supplied at the school by the aid of the Tuberculosis Society. It is an interesting fact that from the twelve schools having ozone all of last year in which about 16,000 children attended there were but three children sent to the open air schools. These schools with ozone are in the more unsanitary and congested districts and yet they supplied but 3 children whereas the

nominal quota would have been at least 25. While not conclusive, all the data points to the probability that as the school system becomes fully equipped with ozone and good air washers the population of the open air schools will disappear. There is no evidence anywhere that the low temperature of the open air school is of any advantage and the tests in the St. Louis Schools prove that the air as now treated is better in every respect than the outside air. The advantages of the open air schools as now administered lies in the diet and rest program. Let provision be made in the regular school for the meals and provide cots for rest and the symptoms of mal-nutrition should quickly disappear.

The writer has had a much more important viewpoint on the use of ozone in ventilation than that stated in the foregoing paper. Evidence was coming from every quarter that blast fan ventilation was losing ground. Many said that with the rising cost of fuel it was too expensive to operate. Split systems were installed because the law required ventilating apparatus but the fans were all standing idle. There was and is a widespread feeling that ventilation is a luxury that must be omitted when the funds are low. It is astonishing how much financial interest can mold sentiment. It was to save the day for ventilation that the present aggressive action was taken to design high grade heating and ventilating plants that should cost no more to build and less to operate than the plain steam radiator plant. This has been done with all the other advantages thrown in for good measure.

The ozone in destroying the odors and other objectionable qualities of the air permitted the recirculation of a large portion of it, which permitted the conservation of the heat which it contained. An examination of the air leaving the building by the old vent stacks proved to be cleaner and better in every respect than the fresh air coming into the building, and the rewashing and repurifying in the ozone machine was all superior to any conditions to be found anywhere else.

The annual coal consumption in a great system of public schools is not subject to influence by personal opinion or to any kind of propaganda. The records of the St. Louis schools show that in the year 1916 there were 115 standard school buildings beside 123 other special and portable buildings and the total coal consumed was 36,400 tons. By 1919 the regular grade schools had increased by nine and one large high school was built and the coal consumption was 32,062 tons. In 1920 the number of large buildings had grown to 128 and the coal consumption for the year was 28,572 tons. Considering the increase of buildings and using present coal price the economy in 1920 over 1916 is about $40,000.00 per year.

Recirculation with ozone has been installed in fifteen old grade schools, three new grade schools and two old high schools containing approximately 25,000 persons. The Hygiene Department consisting of Dr. James Stewart and twelve physicians and 25 nurses supervise all matters very strictly and not a suspicion of injury attributable to ozone has been reported. An enthusiastic principal said recently he considered it almost a crime that ozone can not be introduced into all schools in the face of the St. Louis developments.

The ozone apparatus is purchased and installed in open lettings with

other heating apparatus on specifications requiring results, only, and no particular manufacture mentioned. The contracts have been let to four different firms and always to the lowest bidder. There are at least four manufacturers producing apparatus for this purpose.

TABLE 1. REPORT OF CASES OF SICKNESS IN MULLANPHY AND SHAW SCHOOL BUILDINGS, ST. LOUIS

Causes of Sickness	Mullanphy School Pupils Absent	Mullanphy School Days Absent	Shaw School Pupils Absent	Shaw School Days Absent
Tonsilitis	13	65½	57	220
Sore Throat	24	68½	60	99
Cold	46	264	64	212
Head Ache	9	8½	66	92
Stomach Ache	..	..	25	40
Back Ache	..	..	8	11
Ear Ache	1	1	15	33
Tooth Ache	..	..	20	34
Indigestion	..	..	9	12
Fever	1	21	42	48
Chicken Pox	..	..	1	5
La Grippe	..	..	6	25
Pneumonia	..	..	4	104
Pain in side	..	..	5	9
Diphtheria	1	11	..	..
Dizziness	2	2	..	..
Heart	1	10	..	..
Total	100	475	392	1098
Total Enrollment	1050		1300	

NOTE: Mullanphy School equipped with ozone apparatus, and the Shaw School ventilated according to St. Louis standards and selected as a check.

NO 1 TEST DATA

STREET Nº 4221 SHAW AVE. ST. LOUIS, MO. DATE APRIL 14, 1921

BLDG MULLANPHY SCHOOL ROOM Nº 3 FLOOR 2 TIME 11 A.M. TO 12 M.

STATION	TEMPERATURE DRY BULB	TEMPERATURE WET BULB	R H %	AIR MOTION	DUST	BACTERIA	ODORS	CO_2	SUPPLY REGISTERS AREA ϕ	SUPPLY REGISTERS VELOCITY	SUPPLY REGISTERS C.F.M.	EXHAUST REGISTERS AREA ϕ	EXHAUST REGISTERS VELOCITY	EXHAUST REGISTERS C.F.M.
1	73	63	58	35	5600	12	90	7.9	336	257	864	2.39	315	752
2	73	64	62	35	6750	12	90	6.9	336	215	723	1.21	370	448
3	73	64	62	35	7390	10	90	7.3						
4	74	64	58	35	9820	15	90	6.9						
5								6.0						
6	73	64	62	35	7390	12	90	7.0			1587			1200
7														
8	TEMP AT INLET REGISTER = 71° DRY BULB & 61° WET BULB.													
9	TEMP AT THERMOSTAT = 72°													
10	CO_2 IN OUTDOOR AIR = 3.8 PARTS PER 10,000													

PRIMARY SENSE IMPRESSION WARM NOTES FAINT ODOR WAS NOTICED ON ENTERING THE ROOM. OZONE EQUIPTMENT WAS NOT IN OPERATION.

Nº OCCUPANTS	53	WINDOWS Nº	5	RADIATORS TYPE	SQ FT
PHYSICAL STATE	REST	TYPE SLIDING SASH		NO DIRECT	
AIR SPACE PER OCCUPANT CU.FT.	174	AREA 32 SQ.FT.	176	RADIATION	
TOTAL AIR SUPPLY BY CO_2 PER MIN	1655	LEAKAGE SLIGHT		IN	
AIR SUPPLY PER OCCUPANT BY CO_2 " "	31.2	RATIO TO CU. CONTENT	1:52	ROOM	
" " " " " FAN " "	30	FLOOR AREA PER PERSON	14.5		
AIR DISTRIBUTION %	95				

WEATHER CLEAR
TEMP DRY 59 WET 54½ H.T.
WIND DIR VEL MI PER HR
CHICAGO COMMISSION ON VENTILATION
TEST BY L.A.S. - O.W.A.
PLOTTED BY O.W.A.
APPROVED BY O.W.A.
DRAWING Nº

FIG. 1. RESULTS OF TEST NO. 1 CONDUCTED AT THE MULLANPHY SCHOOL—OZONE EQUIPMENT NOT OPERATING.

TEST DATA

NO 2
STREET NO 4221 SHAW AVE. ST. LOUIS MO.
DATE APRIL 14, 1921
BLDG MULLANPHY SCHOOL ROOM No 3 FLOOR 2
TIME 2^00 P.M. TO 3^00 P.M.

STATION	TEMPERATURE DRY BULB	TEMPERATURE WET BULB	R.H.%	AIR MOTION	DUST	BACTERIA	ODORS	CO_2	SUPPLY REGISTERS AREA ⌀	SUPPLY REGISTERS VELOCITY	SUPPLY REGISTERS C.F.M.	EXHAUST REGISTERS AREA ⌀	EXHAUST REGISTERS VELOCITY	EXHAUST REGISTERS C F M
1	70	62	64	35	6070	4	95	7.1	336	280	940	2.39	300	717
2	70	62	64	35	8580	2	95	85	336	359	1210	1.21	363	442
3	71	62	65	35	6880	10	95	7.6						
4	70	62	64	35		8	95	7.6						
5								72						
6	70	62	64	35	7170	6	95	7.6			2150			1159
7														
8														
9														
10														

PRIMARY SENSE IMPRESSION WARM NOTES A FAINT TRACE OF OZONE WAS NOTICED UPON ENTERING THE ROOM. OZONE EQUIPTMENT WAS IN OPERATION DURING THIS TEST.

Nº OCCUPANTS	57	WINDOWS Nº	5	RADIATORS TYPE	RADIATORS SQ FT
PHYSICAL STATE	REST	TYPE SLIDING SASH			
AIR SPACE PER OCCUPANT	162	AREA SQ.FT.	176	NO DIRECT	
TOTAL AIR SUPPLY BY CO_2 PER MIN	1500	LEAKAGE SLIGHT		RADIATION	
AIR SUPPLY PER OCCUPANT BY CO_2 " "	264	RATIO TO CU. CONTENT	1:52	IN	
" " " " " FAN " "	317	FLOOR AREA PER PERSON	134	ROOM	
AIR DISTRIBUTION %	95%				

WEATHER CLEAR
TEMP DRY 59 WET 54½ H×75
WIND DIR VEL Mi Per Hr
CHICAGO COMMISSION ON VENTILATION
TEST BY L.W.S. - O.W.A.
PLOTTED BY O.W.A.
APPROVED BY O.W.A.
DRAWING Nº

FIG. 2. RESULTS OF TEST NO. 2 CONDUCTED AT MULLANPHY SCHOOL—OZONE EQUIPMENT IN OPERATION.

There may be some who are not fully convinced of the merit of ozone in ventilation. To such an invitation is extended to visit the schools of St. Louis and make such tests as may seem desirable where the facilities for such are so large. It is no longer a matter of opinion of the writer or anyone else. The truth has struck root and must prevail.

TEST DATA

NO. 3
STREET Nº 4221 SHAW AVE. ST. LOUIS, Mo.
DATE APRIL, 19, 1921
BLDG MULLANPHY SCHOOL ROOM Nº 3 FLOOR 2
TIME 10^00 A.M. TO 11^30 AM

STATION	TEMPERATURE DRY BULB	TEMPERATURE WET BULB	R.H.%	AIR MOTION	DUST	BACTERIA	ODORS	CO_2	SUPPLY REGISTERS AREA ⌀	SUPPLY REGISTERS VELOCITY	SUPPLY REGISTERS C.F.M.	EXHAUST REGISTERS AREA ⌀	EXHAUST REGISTERS VELOCITY	EXHAUST REGISTERS C F M
1	70	61	59	35	5400	4	100	7.5	6.1	249	1518	3.7	224	829
2	69	59	55	35	4080	2	100	9.2				1.4	2.14	300
3	68	60	62	35	6150	17	100	7.0						
4	68	60	62	35	6900	8	100	7.5						
5					7010	7		6.4						
6	SAMPLES 5-6-7-8 WERE				8600	10		8.6						
7	OBTAINED IN THE				4950	15								
8	AFTERNOON					3		7.4						
9						*10								
10	69	60	59	35	6151	8.2	100	7.7			1518			1129

PRIMARY SENSE IMPRESSION GOOD NOTES *BACTERIA PLATE NO. 9 EXPOSED THREE MINUTES, APPROXIMATELY 64% OF AIR WAS BEING RECIRCULATED. SAMPLES 1-2-3-4 WERE OBTAINED IN THE MORNING.

Nº OCCUPANTS	49	WINDOWS Nº	5	RADIATORS TYPE	RADIATORS SQ FT
PHYSICAL STATE	REST	TYPE SLIDING SASH			
AIR SPACE PER OCCUPANT	188	AREA SQ.FT.	176	NO DIRECT	
TOTAL AIR SUPPLY BY CO_2 PER MIN	12.58	LEAKAGE SLIGHT		RADIATION	
AIR SUPPLY PER OCCUPANT BY CO_2 " "	257	RATIO TO CU. CONTENT	1:52	IN	
" " " " " FAN " "	310	FLOOR AREA PER PERSON	15.6	ROOM	
AIR DISTRIBUTION %	91.6				

WEATHER CLEAR
TEMP DRY 70 WET 55 RH 38
WIND DIR E VEL 8 Mi Per Hr
CHICAGO COMMISSION ON VENTILATION
TEST BY E.S.H. - E.L.S. - A.L.R.
PLOTTED BY O.W.A.
APPROVED BY O.W.A.
DRAWING Nº

FIG. 3. RESULTS OF TEST NO. 3 CONDUCTED AT THE MULLANPHY SCHOOL—OZONE EQUIPMENT IN OPERATION.

DISCUSSION

F. R. STILL. I do not want to appear as being in opposition to the use of ozone, because I am not. At the same time I do not think we know enough about it to say whether or not it is a thing to be advocated by the Society. Our Research Laboratory is making investigations to learn what we can about it, what merit it has, if any, and how it can be used and what quantity is harmless but effective.

I have here an article which appeared on January 21 in the *Detroit Free Press*, which rather upsets the theory of ozone, if this article can be relied on. It is a report which has been sent out by Dr. Benjamin Moore, of the British National Institution for Medical Research which reads as follows:

NO OZONE NEAR EARTH, BELIEF

London, Jan. 21.—Science has destroyed so many accepted theories that the main worry to the public is to find some pronounceable label with which to distinguish the latest discovery. It will not have to apply its wits to devising another popular name for what it has hitherto understood to be ozone.

Ozone, one has always been instructed, is the substance causing that clean, pleasant smell in fresh air; but Dr. Benjamin Moore, of the British National Institution for Medical Research, has conclusively proved that the latter is nitrogen peroxide.

SAYS IT'S FIVE MILES UP

"There is no ozone close to the earth," he says. "You have to go at least five miles high to find it. That which has been described as ozone is, in reality, nitrogen peroxide."

This and other discoveries have been made by Dr. Moore and his colleagues during experiments extending over eight years, in which they have examined the origin of life.

"Peroxide of nitrogen," he explained, "was believed to be produced by the electricity of occasional thunderstorms; but we have found that strong light is the agent. We found that if you examine rain or dew freshly fallen you will always get this compound in it; let the water stand and it oxidizes into nitrates; expose that to light again and it returns to peroxide.

A GAS DISINFECTANT

"We are now producing nitrogen peroxide in the laboratory, using a powerful mercury light. At present the gas, diluted to 1-50,000 is being employed experimentally at two of the chief London hospitals for disinfecting rooms and clearing bacteria from the nasal passages.

"For the gas is a disinfectant, and it is curious to reflect that the sun, which is itself the most powerful physical disinfectant, should generate another chemical one in nitrogen peroxide.

"If such or similar experiments can be carried out successfully, their value in affording protection against air-born diseases such as influenza and common colds by disinfecting the air of places of assembly is obvious, and it is along such lines that discovery in future will probably appear."

EXPERIMENTS IN COUNTRY

Dr. Moore conducted certain of his experiments in the country air, using chemical mixtures which were responsive to ozone and nitrogen peroxide responsive to ozone and nitrogen peroxide respectively. He found the latter present, but not the former, thus establishing what the present Lord Raleigh had discovered some years ago by projecting a beam of light across a valley and photographing the spectrum—that ozone was absent.

He is convinced that the origin of life is to be found in the action of the sun's rays. He conceives that the primeval world was a brown mass, without sign of life, whether of man, beast, or plant, and that the teeming existence of today is the outcome of the sun's energizing light.

In this connection he carried out an experiment with seaweed at the Marine Biological station at Fort Erin, England.

The Author. The statements which Mr. Still has just read are very similar to many that have been made in the past few years. The ordinary, or I might say the regular chemist is not equipped to do analytical work in this peculiar field of research. Special apparatus is required and new methods must be originated to make these determinations of ozone and nitrogen compounds in the low concentrations found in nature. There is now no doubt of the presence of both ozone and nitrogen peroxide in nature and in all air exposed to an open arc lamp, and following electrical storms in the open. Mr. Armspach and Mr. Yant of the Research Laboratory are reading a paper at this meeting in which they answer this question very definitely. They have developed the apparatus and have made determinations on the two kinds of ozone machines specialized for ventilation, one of which I have used extensively and the nitrogen compounds are found to be but a trace. This agrees with the findings of Professor Anderegg of Purdue University which I had before proceeding in our work at St. Louis.

A Member. I should like to ask what percentage of recirculated air gives the most favorable conditions.

The Author. The amount of air that can be recirculated depends upon the building. If the leakage is great the amount of new air must be larger, as it is necessary to maintain a pressure in the building. In our new schools we can recirculate as much as 85 per cent. Sixty-five per cent was recirculated in the Mullanphy School.

William J. Baldwin. I would like to ask Mr. Hallett what becomes of the CO_2 when the air is recirculated many times. It must increase in the ratio of about three times the disappearance of the oxygen, and the presumption is that the ozone is to take the place of oxygen, so the air can be enriched with something that is not oxygen; but with something that is near oxygen, and this being true, the air being recirculated, the increase of the CO_2 must go on.

I remember, in this society, some years ago being on the platform and getting ready to speak. I was sitting alongside of an ozone machine and when I stood up, my voice was gone. I could not say much and I had to make an apology, but at two o'clock I was all right again. I had been out for an hour and got the huskiness out of my throat. My presumption was, that the ozone was too active and created an inflammation of the mucous membrane of the throat.

However, there are two questions involved. One is, will the ozone repair the air of the room in proper quantities as oxygen would; and the other is, is it not a fact that the CO_2 is continually increasing in the air of the room, making an artificial atmosphere of unbalanced composition unless fresh air is admitted in volume sufficient to keep the CO_2 at some reasonable limit of safety and that the oxygen, or a substitute for it, will also *repair* the air, to a desirable *standard of purity* for continuous breathing?

THE AUTHOR. Our tests as shown on the card indicate that CO_2 runs to 8 or 9 parts in 10,000. Now 8 or 9 parts in 10,000 would not be enough to make any difference in a building. There never could accumulate enough CO_2 to be of any consequence whatever. CO_2 itself has no effect, any more than pure nitrogen has. It is inert. It cannot have anything to do except dilute the air. And since our lungs themselves contain 400 parts in 10,000 there is no chance whatever that an accumulation of the CO_2 in the air would ever be of any consequence at all.

The fact that Mr. Baldwin's throat became husky while he was sitting by the side of an ozone machine is very well understood. We have had a great many cases like that. Had he continued with ozone for two days his throat would have cleared up, even though he remained in the ozone. That condition is only a temporary condition. Any individual trouble for two or three days has been with concentration very much higher than we use in school ventilation. We do not use concentration so high that you can smell the ozone when you are in the room. But we have had many cases like yours and the sensation all passes away and there is no harm whatever. You need not be afraid of any inflammation of the mucous membrane ever setting up from that cause.

P. J. DOUGHERTY. In the development of the ozone question I wonder if it has reached the stage of chemical equation yet. In other words, I would like to know if there has anything developed so far to give us the chemical reactions that take place in the use of ozone. In other words, we know what the lungs exhale—a certain percentage of moisture, CO_2, which in my opinion is nothing more than a diluent, and has no pathological effect, and principally animal matter. Now in talking of that animal matter, what is the molecule that comes from the lungs, and in the second place, what is the chemical action with the ozone?

W. P. YANT. Carbon dioxide is not alone a diluent, but it also has a stimulating effect on the respiration. If an atmosphere is made up keeping the oxygen at 20.9 per cent, which is the percentage found in the atmosphere, and a great excess of carbon dioxide is added it will result in heavy breathing. We have tried out 10 per cent of carbon dioxide in 90 per cent oxygen for resuscitation in cases of carbon monoxide poisoning, the CO_2 being added for the purpose of stimulating breathing, which it gave evidence of doing. From our standpoint, therefore, the fact that the carbon dioxide is alone a diluent will have to be discarded, because it is the natural stimulant for breathing. If pure oxygen is inhaled for a while it will result in a slowing up of the respiration, for the reason that there is not sufficient carbon dioxide in the lungs. But regarding the carbon dioxide which is found in the ordinary schoolroom, if 60 per cent of the air is recirculated, I do not think that it will ever get up to the point that will cause much excessive breathing, because I suppose that limit is about 200 parts in 10,000.

As to the question regarding the chemical action of ozone, as far as we know, it has merely a straight oxidizing effect, similar to that of potassium permanganate. Oxygen or hydrogen in the atomic form is considerably more active than in the molecular form. When O_3 breaks down to O_2 one atom of oxygen is in its atomic form and two in the

molecular form, and the atomic form is a good deal more active than the molecular.

The Author. In regard to Mr. Dougherty's question, there is no doubt at all that the oxidation of the secretions in the membrane have very much to do with what we have been attributing to ozone, producing compounds that are the result of oxygen combining with the secretion. What we get of the reaction there would depend on what the secretion was, of course, and I think a chemist would have no trouble in making a determination as to what your reaction was.

Ozone is not a new kind of chemical, and when it does its work it oxidizes. As to its result in the lungs, I know it raises the hemoglobin. It does not go in as ozone, it goes in as oxygen, in a more active form for entering the blood corpuscles. So I think you can disregard all other things and consider it as merely oxygen, but under favorable chemical conditions.

Wm. J. Baldwin. I would like to ask Mr. Hallett why he adds ozone, rather than circulate the fresh air more freely? Is it to save the heat and lessen the cost of operation, or is there some other advantage?

The Author. That is my reason for it, that is to save heat. I am not considering the medical effect. I am not a doctor and I am not interested in that end of it at all. I am only interested in making the air fit when it is recirculated to go back and give satisfactory results.

E. Vernon Hill. I was very much interested in Mr. Hallett's paper, particularly in his suggestion that instead of speaking of recirculating air he says repurifying air. I would suggest that he go a step further and instead of saying recirculated, purified, ozonized, he should say repurified, recirculated and sterilized air. There is, I believe, more feeling against the use of ozone than there is against recirculation. There are lots of reasons why that is so. Some of them are faulty, some of them apparently not. I think perhaps most of the severe criticism, most of the questions that have arisen, are questions of concentration. I have letters coming to my desk, two or three a week, one will say that they are using ozone in a bank building and the employees complain of a headache and want to know if it is objectionable or not. The next letter will come and say that every one is pleased with the use of ozone and they think it ought to be used more extensively.

I just want to give an illustration. I put ozone in my bedroom last August and have been using it there ever since as an experimental proposition. In the summertime when I had the window wide open and the concentration of the ozone was very low, there was, as far as I could see, but beneficial results. In fact, I gained in weight. In the fall when I closed the window down and it was open only three or four inches the concentration began to get too high and I would wake up in the morning with a headache. So I came to the conclusion that it was entirely a matter of concentration. We have been doing some experimental work in the concentration. We have found that one one-hundredth part in a million of ozone can be detected in the air. That is a very small amount of ozone. And the idea that this concentration of ozone will kill bacteria

is in my judgment erroneous. We have exposed bacteria on culture plates, glass rods, etc., in our institutions up to two hundred parts in a million before we got any decided effect.

Now I think this should be perfectly understood before we go any further in the ozone proposition at all. It has, to my mind, a valuable deodorizing action. The question of whether it masks the odor or whether it actually destroys the odor depends upon the odor and upon the concentration. If the odor is oxidizable and the concentration is sufficient it will destroy it. If the oxidization is low it will act as a masking agent and when the ozone is removed the odor will return. But suppose it is even a masking agent in a low concentration state it has a therapeutic value, for the reason that if you have to stay in a room where objectionable odors are present the tendency is to breathe less deeply than otherwise, and if you have objectionable odors there is a condition of stagnation of blood and consequent injurious effect on the health. So even the masking effect of ozone is a desirable thing as there are no injurious effects in its use. So far as we have been able to learn to date and from anything in literature that I have studied, there is not one iota of evidence that it is injurious in those low concentrations. On the other hand, I have found nothing in my experiments or in the literature to justify the belief that it is going to rejuvenate old men or turn gray hair black or do anything of that kind to improve the health. It is a question, of course, that requires a lot of study and very careful experimental work. It has its value, I believe, if used carefully and conscientiously; but I think our Society should adopt it with open minds and await developments. We certainly should not condemn any more than we should approve.

Mr. Still's quotation that he read interests me. I don't know who the author was, but I have made quite a careful study of the literature of the science for the past ten years from at least thirty different scientists, and there are those who have found high concentration of ozone at the seashore, in the pines, etc., and there are many careful observers who made tests and failed to find it. Then there are more who have found it, and without any question shown conclusively that it is not peroxide of nitrogen or any combination.

WM. LOHMAN. I want to cite an experiment that was made in a cold storage room. The object of using ozone in cold storage rooms is to save refrigeration. There was a question as to whether the eggs that were stored in the rooms benefited by the use of the ozone or not, therefore two rooms were set aside in Glenwood Springs, Colorado, for Swift & Company and a thousand cases of eggs were placed in each one of those rooms. The temperature and humidity in each one of those rooms were identically the same and one room was equipped with ozone, the other was not. The eggs were purchased in Kansas, shipped to Glenwood Springs, Colorado, and one thousand cases were closely candled and examined for spots and rots and placed in each one of those rooms. They were put in the room late in May, or rather the first part of June. The eggs were not removed until January. They were again closely candled, and in the rooms where they had the ozone machine in opera-

tion they found a loss from spots and rots and shrinks of less than three cases of eggs, and in the room where they did not have the ozone machine there were twenty-five cases lost from spots and rots.

In addition to this, in the year 1914 I had occasion to make an installation of a poultry feeding station where all the air was recirculated. There were 67,000 chickens in this room. The experiment took place for several years. It was made by the Fairmont Creamery Company, of Omaha, Nebraska. They have since installed ozone and recirculate their air for all their poultry feeding stations and have found that they have reduced the death rate of their chickens, and they can fatten and feed their chickens faster and in a less number of days prior to killing, with better results than they have ever done before. In Sioux City, Iowa, there are two poultry feeding places, one absolutely next door to the other, one equipped with ozone and recirculated air and the other one that is not equipped with ozone. In the one in which the ozone machine is installed they had practically no death rate, and in the other one they were carrying them out of there every day.

J. M. Robb (written): The basis for this discussion is 15 months' experience with more than 150 electrozone machines, in 10 different towns, covering a great variety of ventilating conditions, preceded by 30 years in heating and ventilating practice.

During the past two years, every opportunity has been sought by study, personal observations, discussion and correspondence, to learn everything possible about ozone, and the more the subject is studied, the deeper grows the conviction that, ozone has possibilities for enormous benefit in ventilation, scarcely appreciated by anyone now.

This conviction compels the effort to emphasize some points in Mr. Hallett's paper, and particularly, some of the points brought out in his first paper presented in January, 1920, that the present one confirms, but does not, it would seem, bring out as clearly as their importance deserves. If their importance were grasped, opposition to ozone in ventilation would melt away as the snow does before the spring sun.

In the attempt to emphasize these points, certain comparisons will be used. It is realized that comparisons are odious. But they serve as standards, and often are the best means to quickly bring out in relief certain views offered for examination. It is hoped therefore, that if any feeling of irritation is aroused by the comparisons used, that consideration of the spirit that uses them, will allay it.

When the history of ozone in American ventilation practice is written four names will stand out with great prominence as pioneers, each in different paths. They are, Dr. Milton W. Franklin, James Todd, E. S. Hallett, and Dr. E. Vernon Hill.

Dr. Franklin's work to secure attention to ozone at a time when it was unknown, will command the greatest respect and admiration for the tireless effort he put into it.

Dr. Franklin writes under date of July 8, 1921: "My convictions were based on over ten years of the most intensive research of my own and the study of the results of other investigators. I have never seen

an adverse report on ozone that was not so thoroughly replete with sophisms that it could be picked to pieces and shown to be false. No fair-minded investigator or referee but was thoroughly convinced as to the merits of ozone in ventilation."

James Todd, president of the Sterling Varnish Co., Pittsburgh, Pa., has conducted experiments with ozone on disease for over twenty years. The preface to a book report he published in 1916, covering seven years of experiment states: "We have nothing to sell. . . . . This effort is an attempt to secure the co-operation of the medical, engineering and chemical professions, for the welfare of man, on a scale never before attempted."

As such, this report deserves the attention of every ventilating engineer. Mr. Todd's work does not relate to ventilation, but it shows the vital importance of ozone in the maintenance of health. It also shows some of the difficulties attending the uniform production of ozone by electrical equipment. It should be borne in mind, however, that his work is based on the use of equipment producing higher concentrations of ozone, than that developed especially for ventilating applications, by Mr. Hallett's investigations.

Rideal, Ozone, 1920, seemingly knocks the ground from beneath all statements that ozone has only a psychological effect. He shows that ozone has a germicidal and tonic effect in concentrations safe to use in ventilation. Todd's final report of his work, now in preparation should be studied by every heating and ventilating engineer.

The basis for the just complaint of ventilating engineers, that physicians, whose proper function it is to define what constitutes correct ventilation, because in its final effect it is a matter of public health, refuse their responsibilities, might be eliminated by a close attention to Mr. Todd's work, and a public statement that the ventilating engineers are prepared to perform their part of the co-operation Mr. Todd seeks.

Mr. Hallett's work can be best valued by comparing it to that of the railway master mechanic who rebuilt a wrecked bridge while the chief engineer scurried about to collect draftsmen and assistants to prepare plans and specifications.

Doubt of Mr. Hallett's results has been expressed because of his failure to make proper research, before actually doing what he has done.

The health of those affected must be the final answer to the question, what is correct ventilation? and the answer given by the licensed physicians who pass on the results delivered by Mr. Hallett's application of ozone, must outweigh any laboratory research, no matter how carefully conducted.

To compare Mr. Hallett's work to the rebuilding of a wrecked bridge is no mere figure of speech, for the growing number of costly mechanical ventilation plants that owners refuse to operate, for various reasons, and the increasing number of buildings where mechanical ventilation should be installed, but is omitted, because some one connected with the operation has been disappointed in some ventilating application, indicates a state of affairs, that must cause concern to all really interested in ventilation practice.

A revolution in ventilation ideas is indicated by Mr. Hallett's applica-

tion of mechanical ventilation in a million and a half dollar high school structure, with no vents through the roof, no direct radiation in class rooms and similar spaces, and only a single trunk duct carrying air at constant temperature governed by the maximum heat loss of the worst exposure, with automatic temperature regulation effected by reduced air flow to spaces where the heat losses are less.

The debt the ventilating profession owes to Mr. Hallett for his breadth of view, and for his courage to apply his views in the largest working application so far made with ozone, is no small one.

Dr. E. Vernon Hill's record as an aerologist requires no mention to members of the engineering profession. His attempt to define the principles of ozone application in ventilation as described in the *Heating and Ventilating Magazine* is pioneering of the most valuable sort.

Let it be understood that just as skill based on requisite practice is a prime essential to securing a given heating effect or a given effect in air movement, so equal, if not greater skill is requisite for proper ozonation. The law of diffusion of gases has been presumed upon in the application of ozone, with more failures than have attended the efforts of the unskilled in heating and ventilating applications.

Ozone with all its powerful oxidizing effect, can be as safely used in ventilating applications, as steam is used in heating, if the procedure Mr. Hallett outlines is observed. The power of steam for scalding and burning is well known, yet practice has taught the safe limits within which it can be used, so that it is now recognized as a most useful servant. As useful as steam, with all its latent capacity for harm, has proved to be, ozone has equal possibilities in ventilation.

Those who fear to use ozone may be divided roughly into two classes, those who fear that its use may result in lowered ventilation standards to the extent of requiring less mechanical ventilation and less skill in the application of ventilation. And those who honestly desire to use it if benefits are as represented, but fear possible harm because of lack of experience with it.

No fear could be more baseless than those of the first class.

A proper ozone application is infinitely more than the selection of an apparatus of necessary capacity. The same skill that is required to diffuse light, heat or air over a given area, is required in far greater refinement to secure a uniform application of ozone.

For a given freshening effect in a definite space three entirely different effects will be secured if a single unit of adequate capacity is used and spotted somewhere within the space to be treated; or a multiple of units of a total capacity equal to the single one, is used and the units spotted with the same care that a heating engineer spots his radiators for uniform heating effect, or a lighting engineer spots his lights for equal lighting effect, or if a different type of ozone equipment be used with a central fan and duct system to uniformly diffuse the ozone freshened air over the entire area, *with proper relation between fan capacity and total volume of space to be freshened.*

But the briefest consideration of the above will refute the objections of those who suspect that if ozone is what it is thought to be by its

advocates, a few small brass fans will be all the mechanical ventilation required.

One consideration alone, of many that could be offered, will show how baseless are the fears of those who dread possibilities of harm from any application of ozone in ventilation within the limits Mr. Hallett indicates. That is the widespread use of X-ray machines. These are almost invariably installed in small unventilated enclosures. Their operation makes higher ozone concentrations in these enclosures than would ever be considered in a ventilating application. Many of the operators work as long hours in these enclosures as classes are held in schools, or as clerks work in offices. Yet there seems to be no evidence of harm resulting.

Leonard Hill, considered by many as the foremost physiologist in the world, says (*Scientific American* Supplement, June, 1912): "The effect of ozone in weak concentrations, on the olfactory nerve and those of the skin and respiratory tract, is its justification in ventilation." Todd's and Hallett's work, one experimentally and the other in practical large scale application, prove that ozone has a far more vital relation to health than this.

The use of ozone in English military hospitals (Rideal Ozone 1920, p. 157) in treating open wounds, shows how baseless is the fear of harm from irritation from too high concentrations in ventilation and justifies Mr. Hallett's conclusions as to the cause of any irritation noticed.

Although the nose is as yet the most effective and delicate instrument for judging ozone applications in ventilation, considerable experience is very essential to determine the difference between the real ozone odor and a combination odor that very often results from the first few days of ozone application.

Von Kupfer in discussing ozone at the Congress of Heating & Ventilating Engineers, Frankfort, Germany, 1910, calls attention to this, saying: "Peculiar and disagreeable are the mixed odors for the first few days of ozone application." He advises making the first use of ozone during hours when the building served is not in service, for considerable periods.

All of the 150 machines installed by the writer were put in on 30 days' trial. In a great number, if decision had been made in the first week of operation, rejection would have been certain. In some instances, the machines could not be operated during working hours for the first week. Later, the machines could be operated with no perceptible odor, at full capacity, where during the first few days, a fraction of the total capacity gave a most disagreeable odor.

Though the cause is entirely different, the effect seems to be very similar to the odors of grease, pipe joint cement, paint and varnish, that a new heating plant gives off during the first few days of operation.

Consideration of this will instantly show how much care must be exercised in judging ozone odors, and how unreliable any estimate of concentrations to effect detectable odors may be, without proper regard to the oxidizable content of the air used.

It has been observed also that if customers failed to operate their equipment constantly in accordance with directions, the disagreeable combina-

tion odor persisted continuously until enough ozone had been put into the space to be freshened, to thoroughly sterilize it.

As an indication of how little value tables of ozone concentrations for certain effects, are without some way of determining how much oxidizable matter any given atmosphere to be freshened, may carry, consider two adjacent ozone applications. One is in a department store restaurant kitchen to destroy cooking odors that found their way into the store from a corridor of 300 cu. ft. volume, to a four-story elevator shaft. In this application the ozone generator capacity of 12,000 cu. ft. of air per minute is discharged, with no ozone odor perceptible in the year that the application has been in service. Neither is any cooking odor perceptible in the corridor or elevator shaft.

In an adjoining bank of 500,000 cu. ft. of volume, ozone generators of same type and on same electric service wires, of a capacity of 6,000 cu. ft. of air per minute, give a very pronounced ozone odor if operated to exceed one-half capacity. In this same bank, during the summer when windows are opened, the machines must be operated at full capacity to get any freshening effect. Other summer experiences in offices and stores confirm the instance of benefits from summer use of ozone cited by Mr. Hallett.

In the same manner, laboratory experiment on the ozone concentration required to destroy bacteria on a culture plate, is no indication of the extent of concentration necessary for complete germicidal effect in a ventilating operation.

Ozone is used to speed the drying of printers' ink, paints, varnish, etc., and it might well be that exposing a culture plate to ozonized air would have the effect of preserving the bacteria. The correct way to determine the germicidal effect of ozone in low concentrations permissible in ventilation, is to expose culture plates, under identical conditions, before and after ozonizing.

One of the writer's installations is in a candy factory, where in a space of 18,000 cu. ft., with air rotated at the rate of 2,000 cu. ft. per minute, with construction built like a bank vault, no outside air supply and housing 45 employes occupied 10 hours per day, culture plates before ozonizing showed bacteria colonies too numerous to count. After ozonizing, with a low concentration almost imperceptible to smell, but with a sufficient length of time to sterilize ducts, refrigerating coil housings, etc., culture plates exposed late in the working day, under as nearly identical conditions as before ozonizing, showed sterile after 48 hour incubation. The first exposures, before ozonizing were examined after 24 hours of incubation.

This installation was one where foul combination odors occurred during the first week of operation, at the lowest production at which the electrozone machines could be operated. After the first week, the full capacity of the machines gave very little odor, but a remarkable freshening effect. During the Christmas rush when every operator was speeded to the limit, the forewoman reports the girls stating that they felt less fatigue than when turning out less work before ozoning.

Observations indicate two facts.

First: The mechanical agitation of the air only to be secured by proper ratio between volume of space to be freshened and rate of flow, with due consideration of location of inlet and outlet openings, seems to have a very definite function. The effect seems to be parallel to the findings of water purification engineers, that right proportioning of chemical reagents for removal of certain impurities, are not effective unless thorough agitation is secured. And in addition to agitation, the time factor is not the least important.

Second: The ozone concentration necessary for a given duty cannot be ascertained previous to actual trial on a working scale, until it is definitely known what amount of oxidizable matter the air to be handled contains, and how much ozone is necessary to balance this destruction. In actual application, only the amount of ozone not utilized in the destruction of such matter, is left for any other effect.

The problem is identical to the supply of heat with a blower system, in which to maintain a given room temperature, the air must be blown into the room at a temperature so much higher than the one to be maintained, that the drop caused by heat losses will hold the desired temperature.

Only in ozone application, there is no instrument like the thermometer to indicate results. The nose, however, is a most certain and dependable instrument, to indicate results with ozone. Doubtless, in due time, dependable instruments will be available.

The basis of the application of ozone, is the theory that any atmosphere containing a little excess of ozone, *properly diffused*, so that its odor is barely perceptible, when coming in from out of doors, will be pure and healthful.

But in this connection, the mixed odor resulting from an inadequate, or improperly operated ozone application must be guarded against.

The certain way to guard against deception by this mixed odor, is to operate the equipment for a period of from one day to a week, depending upon age of duct system, character of odor giving material to be destroyed, and a number of other circumstances that vary in each application, such as rate of air movement, location of outlets that may retard diffusion, etc., until it is certain that thorough freshening has been effected

The skilled ventilating engineer need have no hesitation about using ozone with the type of machine developed to meet Mr. Hallett's criticisms. His experience indicates to him the rate of air flow a given set of conditions would require, without the use of ozone. If he will then be guided by the manufacturers' recommendations for capacity for this rate of flow, he will operate within safe limits.

He may find on checking operation, that he will be obliged to operate his equipment a few more hours than he contemplated, to secure the best freshening effect. And he may, in certain odor removing applications, such as gymnasium locker rooms, restaurant kitchens, etc., be obliged to materially increase the ozone producing capacity for the complete destruction of the odors.

But he will have all the assurance he now has in the use of low pressure steam for heating, that he will do no harm.

The experience on which this discussion is based warrants the expression of the conviction, that any engineer who can design ventilating applications that will meet the requirements of the Synthetic Chart, who will arrange for recirculation and ozone application, will find results so satisfactory as to fully substantiate Mr. Hallett's expression that ozone freshened indoor air is superior, *for health maintenance,* to the best out-of-door air he can provide.

But let it be clearly understood, that the results will be proportionate to the skill of the engineer in handling his air. As the chance of an amateur, for success in the design of a comprehensive ventilating application, is a 1000 to 1 shot, so also is the chance for one not skilled in ventilation, to obtain success with ozone.

If the dust hazard in ventilating systems were fully comprehended, no system would be considered safe without an ozone application to insure thorough sterilization of the dust that inevitably settles in every duct system.

Objections have been made, that only irresponsible reformers find any menace in the dust accumulations in ventilating ducts. It is argued that, if this dust were as harmful as it is urged, that part of the population that uses hot air furnaces, would be down and out.

This mention of hot-air furnace practice, overlooks the sterilizing effect of the high temperatures at which hot-air furnaces are operated. And it further overlooks the low health of the best part of our population indicated by draft records. Further investigation may rightfully place a part of the blame for this on the dust accumulations in the universal hot air furnace application, even though the high temperatures carried burn much of this dust to carbon.

But in a ventilating system with air warmed by low temperature surfaces, particularly when the air is humidified, tropical conditions are set up, that increase the danger.

Does any one recognize this quotation? "The heating and ventilating system may have much to do with the spread of disease by bringing germs from the outside air into the room; indeed, what better means for carrying germs can be provided than the central ventilating system for distributing the germs from the fan room throughout the whole building?" Refer to President J. R. Allen's address, Proceedings, A. S. H. & V. E., page 59, Vol. XIX, 1913.

Where duct systems are unavoidable, ozone equipment should be considered essential for sterilization alone.

And it should be quite clearly understood, that ozonation does not necessarily mean recirculation. In one of the applications of this experience, there is an office in which 50 clerks are employed, that receives 6000 cu. ft. of outside filtered air per minute. There is no direct radiation, there is automatic humidity and automatic temperature control. The plant was considered by its owners before ozone was added as the best office ventilation application in their state. The benefits of ozone have been as apparent in this office as any other application of the one hundred and fifty.

Evidence is growing, that the usual mechanical ventilation application

defeats its purpose as effectively as an attempt to purify the water of a tile lined swimming pool, by pumping in great quantities of water from a turbid stream.

If ventilating results were as visible as results in operating a swimming pool, recirculation and ozonation would have become standard practice years ago. It is repulsive, of course, to many minds to think of using air that others have used, water that others have bathed in, or food that others have touched. But, if the facts are faced without dodging, the growing number of theaters, banks, offices and similar institutions that are driven by increasing fuel costs and other considerations, to recirculate, either with or without air washing, will compel the conviction that the only certainty the fastidious have, that they are not breathing air that has also served others, is to dwell permanently in the country.

If these fastidious people would examine ozone freshening carefully, from successful installations, not from the numerous failures, they can not escape the conviction that a proper ozone application carries on nature's cycle of air purification indoors, just as she performs it out-of-doors, when not hindered by man's community processes. And that proper ozonation is the best insurance of pure air they can get in community life.

When theater managers, operating 2500 seat theaters with refrigerated air at a cost of from $500.00 to $700.00 per month, for power and water alone, for four performances daily, find that they can recirculate their entire air in summer, without any one being the wiser, the temptation to do likewise in winter, to save fuel, is too great for human nature to resist. In such cases and others through the entire range of ventilating application, ozone should be used. And it would be, if designers and operators were aware of the insurance of pure air that it carries.

In the efforts that have been made to secure a greater number of comparisons, such as Mr. Hallett and The Hygiene Department that checks his installations, have made in the Shaw and Mullanphy Schools, the question has been asked, how would the money be raised to make such comparisons on a wide scale?

If five interested and earnest engineers in representative cities, where school ventilation standards approximate those maintained in St. Louis, would interest themselves in ozone to the extent of comparing two buildings in their respective cities, one with and one without ozone, for a full winter, just as was done in St. Louis, with absences and causes of absence, no money need be raised.

Of course some agreement would be necessary as to the make of machine to be used, and the manner of application.

And in making the comparison, ozone should be compared with the best type of ventilation available. It would be a waste of time to apply ozone to some faultily designed ventilation application and expect to compare its results with a better designed plant. The two different buildings should be alike in equipment and character of humanity housed Then a winter's results would bring out information that would lead to more certain conclusions.

In one of our most important cities, an ozone application in a public school, with, however, a different type of machine from the one Mr. Hallett uses, has failed to give any appreciable benefits.

Any engineer accustomed to examining the performance of mechanical ventilation of different designers, needs no outside advice to realize how differently such plants, using identical equipment, but in different combination, and varying skill in application, perform.

Such engineers will readily understand that the failure to get results in individual plants may have no bearing whatever on the principles involved.

What this discussion attempts to express, will show to the skilled ventilating engineer the variety of factors that may affect successful ozonation, and how the neglect of proper consideration for any one of them, may make just as much of a failure, as similar neglect has caused in a great percentage of ventilating installations.

CONCLUSIONS

Summed up, the experience back of this discussion seems to warrant the following 15 conclusions:

1. The atmosphere of all communities, even the smallest, is contaminated. (See Rideal's Ozone, 1920, p. 24.) The larger the community, the greater the contamination. As humanity can exist but few minutes without air, while it can endure for days without water, and for weeks without food, it follows that a contaminated air must have a far greater cumulative effect for disaster, on health, than impure food and water, whose dangers are fully recognized. The increase of tuberculosis is but one manifestation of this.

2. Ozone is in no way a substitute for mechanical ventilation. It can not be effectively utilized without adequate mechanical ventilation. And it is a powerful supporter for mechanical ventilation when it is used with adequate skill and understanding.

3. A proper ozone application means the mechanical operation indoors, of nature's natural air purification process out-of-doors, to restore that quality for the best maintenance of a vigorous state of health that contamination has destroyed.

4. It does not necessarily mean recirculation. But, if, during the heating season, especially, the ozone freshened air is superior to out-of-door air for the maintenance of health, as evidence begins to indicate, why incur the cost of heating and cleaning the out-of-door air?

5. A proper ozone application means the maintenance of that concentration of ozone that nature maintains where man's community activities have not destroyed it, *after introducing and diffusing sufficient ozone to destroy all impurities.*

6. This means a rather definite ratio of air movement to volume of space to be freshened. The minimum is certainly not less than the present standard of 30 cu. ft. of outside air per person per minute, and best results, particularly where odors are to be destroyed, probably require a considerable increase above this. In short, what seems to be demanded is less heating capacity, but greater air moving capacity.

7. Diffusion is everything. As extended examination of mechanical ventilation applications show great variation in the skill of designing and erecting them, it follows that there will be found the same variation in ozone freshening results, depending on the skill of those who make the application. It is futile to expect the best results from the application of ozone equipment to inadequate mechanical ventilation.

8. It seems probable that central fan design could profit much by abandoning the low velocity horizontal discharge air inlets, in favor of the high velocity, vertical up discharge, of the unit ventilator, which secures such admirable diffusion by cataracting the incoming air against the ceiling.

9. As ducts with their extensive dirt collecting surfaces, are as unsanitary for handling air, as similar uncleanable conduits are for handling any food material, every effort should be made to utilize the natural passages of the building for

introducing the air for ventilation. Visibility insures cleanliness. The unit ventilator practice of utilizing corridors for waste air flow, might profitably be reversed, for use of such passages to introduce air, and so insure a constant flow of air at room temperature, from center of building towards outside walls and windows.

10. Heat to compensate losses and so maintain temperatures should be carried by the air supplied by ventilation, so that temperature drop would follow insufficient air movement. Present practice of compensating heat losses by raising air above room temperatures, and wasting the air at the room temperature, seems based on wrong conception of principles. Air removal at windows, or through vents beneath them with sufficient rate of flow to keep motion definitely towards them would permit air to be supplied at room temperature, with drop caused by heat losses occurring in conduits returning such air to the central heaters for again raising to room temperatures. The resulting advantages will be apparent.

11. Laboratory data as to ozone concentrations for certain effects is of no greater service in design, than data on heat emission rates of radiators without data to indicate total amount of heat that a given space must have to replace heat losses. Before ozone concentration data can be used in design, means must be devised to ascertain the oxidizable content of the air that is to be freshened.

12. For odor destruction, excess ozone capacity must be provided in the ozone generator, or else the equipment must be operated for a considerable period before and after the use of the space to be freshened, to thoroughly sterilize it. The problem is similar to heating, where the greater the capacity, the shorter will be the time for producing a given temperature rise.

13. In judging the effect of an ozone application, mixed odors must be anticipated and guarded against by operation a sufficient period before operations are to be tested, to insure the thorough oxidation of all matter.

14. In judging the efficiency of an ozone application to destroy bacteria, odors, tobacco smoke, etc., there is but one accurate way: to test for such impurities before and after ozone application, with all conditions identical for such tests.

15. As the ultimate end of ventilation is the preservation of the best state of health, the final test of any method of air freshening such as ozonation, is the effect on the health of those concerned. For judging the comparative performance of different methods of ventilation, mechanically, no better device or means perhaps, will ever be available than the Synthetic Air Chart. But a comfortable air does not necessarily mean an air that is best for the maintenance of the most vigorous state of health. Hospital records show this. The real test is observations on the health of those concerned. And this can be shown only by such comparisons extended over a sufficient period of time to let the results come out, as Mr. Hallett has made.

No. 618

HEAT TRANSFER BY CONDUCTION AND CONVECTION

By W. K. Lewis[1], W. H. McAdams[1] and T. H. Frost[1], Cambridge, Mass.
Non-Members

IN the design of equipment for heating gases or liquids by heat transfer through pipes or other heating surface by steam, hot liquids, or similar sources of heat, the most important and difficult thing is the determination of the coefficient of heat transfer to be used in calculation. Every engineer realizes that in air heaters one obtains in commercial practice coefficients varying from 1 to 15.[2] In steam heaters for heating liquids such as water, heavy acids, oils and the like, coefficients are found as low as 10 and as high as 800. It is absolutely necessary to be able to predict with reasonable accuracy the coefficient for a given case or else one must allow excessive factors of safety which involve an unnecessarily large investment.

This point is well illustrated by the following case. The writers were in contact with a company which wished to install a pipe cooler. The data as to duty and terminal conditions were furnished to a number of manufacturers of such equipment. One of these, a manufacturer of a cooler of modern construction and efficient design, recommended a certain size of equipment. The authors estimated the necessary size of this equipment and decided that a cooler of approximately one third the size recommended by the manufacturer would suffice. The manufacturers not only refused to guarantee the performance of this smaller equipment but strongly recommended against its installation, expressing their conviction that it would not work. However, the smaller equipment was installed and functioned satisfactorily, thereby indicating that the manufacturer was using a factor of safety of approximately 300 per cent in the design of this equipment.

FILM CONCEPT

It is well understood that the most important factor in determining heat transfer coefficients is the velocity of gas or liquid through the apparatus. Thus, it is recognized that high air velocity through an air heater will give high heat transfer. The same is true of high liquid velocity through a liquor heater. The reason for this has been under-

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[2] All the numerical values of coefficients quoted in this paper are given in the same units, namely, B.t.u. per hr. per sq. ft. per deg. fahr.

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stood for three quarters of a century, but the fact and its consequences are even yet not fully appreciated by many engineers.

A liquid or gas in contact with a solid surface moves with relative freedom in the center of the mass but the fluid[1] near the surface moves very much less freely, so much so that one can picture it as a stationary film of liquid or gas on the surface of the solid. All fluids, except the molten metals, are such poor conductors of heat that this stationary film offers a large resistance to heat flow. The coefficient of heat transfer in a given case will depend mainly on the thickness of this film. High velocity of the liquid or gas past the surface thins down this film. Furthermore, it has been demonstrated that it is the weight of fluid passing the surface rather than its linear velocity that determines the film thickness. Consequently, the heat transfer coefficient is determined primarily by the *mass velocity* of the fluid, i. e., the weight of fluid, whether liquid or gas, passing through each sq. ft. of cross section of path per unit time.

RESISTANCE CONCEPT

It is obvious that there will be a film on both sides of the heating surface and that each of these films is independent of the other. The heat must therefore traverse three resistances to heat flow; *first,* the film on the hot side, (condensed water in the case of steam, or stagnant liquid in the case of hot water, hot oil and the like), *second,* the metal of the heating surface wall, and *third,* the film on the cold side, consisting of stagnant gas or liquid. A low overall coefficient of heat transfer may be due to high resistance in any one or more of these three layers. It is obvious, therefore, that these layers must be considered separately. The engineer is loath to do this. He prefers to use an *overall* coefficient, *H.* Where conditions are strictly comparable the use of this coefficient is allowable, but it must be remembered that comparable conditions are seldom found. It is believed that the use of these overall coefficients is the one major cause of the present deplorable state of design of equipment of this general type.

The engineer will probably object that the use of *film coefficients* complicates a problem already sufficiently difficult. It is admitted that a certain complication results, but it is very slight if handled in the proper way. Heat flow should be calculated in terms, not of conductivities as is usual, but of resistances. No engineer calculates the flow of electricity in terms of conductivity—only the chemist does this. Heat flow is proportional to the temperature difference and inversely proportional to the total thermal resistance. The total resistance met in any case in question is the sum of the resistances of the metal wall and of the two films upon it. One has therefore merely to add the three resistances in order to determine the total. The resistance of the metal wall is often negligible as compared to the other two. However, its resistance must not be forgotten because in the case of thick walls, or of metals of low conductivity, or at extremely high rates of heat flow, it may become an important factor. The resistance of the fluid film will obviously be inversely proportional to the thermal conductivity of the fluid, directly to the thickness of the film, and inversely to its area. It is almost impossible

[1] The term *fluid* is used to indicate either a liquid or a gas.

to measure the thickness of the fluid film. It is, however, known that this thickness is nearly inversely proportional to the velocity of the fluid passing the surface, and also directly proportional to the viscosity of the fluid. It is usually most convenient to ignore the unknown thickness of the film and to call the resistance equal to $1/hA$, where A is the surface area of the film and h is the film coefficient of heat transfer, a quantity nearly proportional to the mass velocity of the fluid past the surface, to the absolute heat conductivity of the fluid, and inversely proportional to the viscosity. This last term varies appreciably with temperature for a given liquid, and widely from one liquid to another.

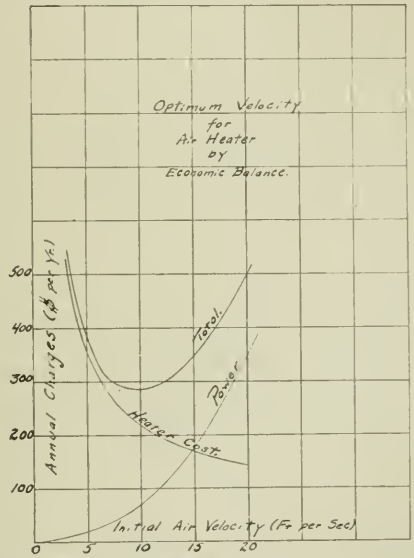

FIG. 1. CURVES SHOWING OPTIMUM VELOCITY FOR PIPE-COIL HEATER BY ECONOMIC BALANCE.

GAS FILM COEFFICIENTS

It not infrequently happens that one of these film resistances is so small that it may be ignored. Thus, in heating air with steam the thermal resistance of the film on the steam side is so low that the resistance on the air side alone need be considered. This is the basis of the formula of Carrier[1] for heat transfer in air heaters with air flowing at right angles to the outside of 1 in. pipes, arranged to give about 50 per cent clear area.

$$H = h = \frac{22.3\ w}{1.42 + w}$$

where w varied from 0.25 to 1.50 lb. per sec. per sq. ft. of minimum clear area, and h varied from 3.31 to 11.5 B.t.u. per hr. per deg. fahr. per sq. ft.

There has been developed by H. C. Weber[2] a somewhat similar formula,

All symbols are defined in the nomenclature table at the end of the paper.

[1] W. H. Carrier, Trans. A.S.M.E., Vol. 33, (1911).
[2] H. C. Weber, Undergraduate Thesis, Chem. Eng. Dept., Mass. Inst. of Tech

which is, however, much more general. This applies to any gas flowing *inside* pipes in turbulent motion. The value of h obtained from this formula is the heat transfer coefficient of the gas film, not the overall coefficient.

$$h = \frac{0.88\ v^{0.8}\ C_p \sqrt{T}\ S^{0.2}}{M^{0.3}}$$

The formula of Weber is based on a critical comparison of the data of Nusselt, Josse, Jordan, Fessenden, Babcock & Wilcox and others. In form it is very similar to that of Nusselt. This equation was derived from data for the flow of carbon dioxide, air, and illuminating gas in pipes ranging from ½ in. to 2 in. in diameter, at mass velocities from 1 to 20 lb. per sec. per sq. ft. transverse area of tube, and for temperatures from 70 to 2000 deg. fahr. Comparisons of this formula with isolated tests on various sizes of large pipe, up to 12 in., indicate that the equation may be used for extrapolation with safety, the observed values on such large sizes being somewhat higher than the predicted.

For any given gas, this general formula can be much simplified. Thus for air at ordinary temperatures, in 1 in. to 2 in. pipes, $h = f\ v^{0.8}$, where f varies from 3.7 to 4.5, (4.2 for air at an average temperature of 200 deg. fahr. in a 1 in. pipe).

It is an interesting fact that this formula derived from data on gases and designed solely for use with gases is not entirely inapplicable to liquid films. This formula contains a viscosity factor although it does not appear as such, as some of its terms are proportional to gas viscosity. Weber's formula should, then, be applicable to a liquid, the viscosity of which is not too greatly different from that of the gases. Thus Clement and Garland found a water film coefficient in a 0.985 in. inside diameter pipe of approximately 1420 at a water velocity of approximately 5.33 ft. per sec., the mean temperature being 83 deg. fahr. For this mass velocity of water, Weber's formula would give a heat transfer coefficient of 1950. Note that whereas Weber's formula was derived from data based on gases, in most cases giving heat transfers from 2 to 20, it is here used with moderate success in extrapolation up to heat transfers of 1950. Such agreement demonstrates that this formula adequately indicates the influence of mass velocity on heat transfer.

For gases *outside* and at right angles to staggered pipes the writers employ a modified form of Weber's formula[1], namely,

$$h = \frac{3.28\ v^{0.8}\ C_p\ \sqrt{T}\ S^{0.2}}{M^{0.3}}$$

This equation has been tested only on the flow of air.

LIQUIDS FLOWING THROUGH PIPES

The following formula[2] has been determined for liquids flowing in pipes:

$$h = \frac{(.112)\ c\ k\ u^{0.8}}{z\ m^{0.17}}$$

[1] Walker, Lewis and McAdams, Chem. Eng. Notes, M.I.T.
[2] Frost, M. S. Thesis, M.I.T., 1921.

This holds for a range of pipe diameters varying from ½ in. to 2 in. and also in annular spaces between 2 in. by 3 in. pipe and several somewhat smaller combinations.

The velocity range is from 1 to 15 ft. per second, and h varies from about 200 to 2000.

The recalculated data of Stanton[1] on smooth drawn copper indicate values of c of about 1.0 while those of Frost and Manley[2] in ordinary standard steel pipe give 0.50. The work of Stanton is checked fairly well by the recalculated data of Clement and Garland[3] in *smooth* 1 in. cold drawn Shelby steel tubing.

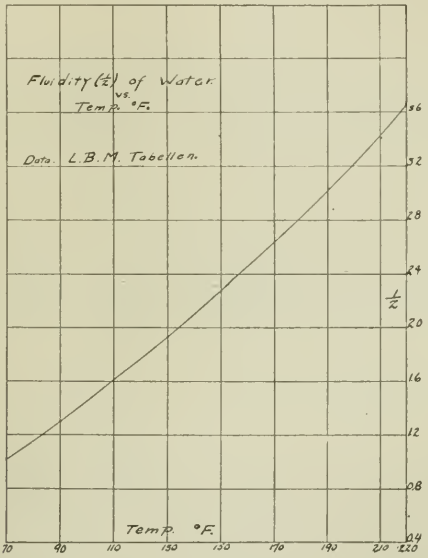

FIG. 2. CURVE SHOWING FLUIDITY OF WATER AT VARIOUS TEMPERATURES.

Since the thermal conductivity of water is 0.329 and the hydraulic radius correction is relatively small, the above reduces, *in the case of water,* to the form:

$$h = 400 \frac{c\, u^{0.8}}{s}$$

the values of c being taken as given above. For the present, it is suggested that c be taken as 0.50 for any metal tube. This equation has been tested only on water and mineral oil. In the few cases where it has been tried with oils, it gave satisfactory checks with the observed values, and hence, in view of the lack of further data, it is suggested that it be used for oils.

[1] Stanton, Phil. Trans. Vol. 190 (A), p. 67, (1897).
[2] Frost and Manley, Chem. Eng. Thesis, M.I.T., 1921.
[3] Clement and Garland, Univ. of Ill. Eng. Expt. Sta., Bull. 40, (May, 1909).

However, it should be noted that this equation will not apply below the critical velocity which is figured by the following equation $u_c = \frac{y\,z}{D\,s}$, where y may vary from 0.12 to 0.36, depending on whether or not unstable viscous (non-turbulent) motion obtains. Obviously the heat transfer apparatus should be designed to operate at a velocity well above the critical velocity, for if it were lower there would be no turbulence or convection to assist the process of heat transfer by conduction. For example, the critical velocity in a 2.07 in. pipe of an oil having a viscosity of 50 times that of water, a sp. gr. of 0.9, would vary from 3.22 to 9.66 ft. per sec.

CONDENSING VAPOR FILMS

McAdams and Frost[1] have determined the following equation for condensing vapors in the absence of any appreciable percentage of non-condensable gas: $h = 2200 \frac{k}{z}$. Experimentation covered steam and carbon tetrachloride. Its use is recommended for the present, although it should be understood that additional work is being done to determine the effect of vapor velocity and hydraulic radius on this coefficient.

Since k for water is 0.329, this equation becomes, for condensing steam, $h = \frac{725}{z}$, where z is the viscosity of the condensate on the tube surface.

In the case of condensing steam containing air, the equation offered by Robinson[2] may be used, until the results of investigations now in progress are available, that is, $\log_{10} h = 1 + 0.0246\,x$, where x varied from 67 to 99.7 per cent steam by volume. In this equation it should be noted that the percentage of steam is considered as the only variable, while it is admitted that other variables such as velocity would have an effect. Work is in progress to clear up this point.

DESIGN OF CONTINUOUS WATER HEATER

A continuous water heater is to be designed to warm 30,000 lb. of water per hour from 70 to 170 deg. fahr., using steam condensing at 220 deg. fahr. outside copper tubing having an actual inside diameter of 5/8 in. and a wall thickness of 1/16 in. The thermal conductivity of the copper is 220 B.t.u. per hour per sq. ft. per deg. fahr. per ft. of thickness. The following calculations are based on a linear water velocity of 4 ft. per sec.

Solution:

a. Heat transfer rate = (30,000) (1) (170-70) = 3,000,000 B.t.u. per hr.

[1] McAdams & Frost, *Journal Ind. Eng. Chem.* Vol. 13, No. 1 (Jan., 1922).
[2] Clark S. Robinson, *Journal Ind. Eng. Chem.* Vol. 12, No. 7, p. 644 (July, 1920) and Vol. 12, No. 8, p. 818 (Aug. 1920).

b. Temperature difference, steam to water:

Initial $= 220 - 70 = 150$ deg. fahr.

Final $= 220 - 170 = 50$ deg. fahr.

$$\text{Mean} = \frac{150 - 50}{(2.3)\ (\log_{10} \frac{150}{50})} = 91 \text{ deg. fahr.}$$

c. Overall resistance (R) to heat transfer:

$$R = r_1 + r_2 + r_3 = \frac{1}{h_1 A_1} + \frac{L}{k A_2} + \frac{1}{h_3 A_3}$$

h_1 (film coefficient steam to pipe surface) $= \dfrac{725}{z}$.

Outer surface condensate $= 220$ deg. fahr.

Inner surface condensate $= 190$ deg fahr. (estimated).

Mean temperature condensate $= \dfrac{410}{2} = 205$ deg. fahr.

From Fig. 2, corresponding value of $\dfrac{1}{z} = 3.3$. Then

$h_1 = (725)\ (3.3) = 2392.5$

$$r_1 = \frac{1}{(2390)\ (3.14)\ (0.75)\ (N)} = \frac{0.00213}{N}$$

where N represents the total linear feet of copper pipe.

$$r_2 = \frac{L}{kA_2} = \frac{0.0625}{(12)\ (220)\ (3.14)\ (0.688)\ (N)} = \frac{0.000131}{N}$$

$$h_w = 400\ \frac{c\ u^{0.8}}{z}$$

Value of $\dfrac{1}{z}$ at mean water temperature of 120 deg. fahr. is 1.82 (Fig. 2). Taking c as 0.5, as recommended above,

$h = (400)\ (0.5)\ (4)^{0.8}\ (1.82) = 1103.6$

$$r_3 = \frac{1}{hA_3} = \frac{1}{(1215)\ (3.14)\ (0.625)\ (N)} = \frac{0.00503}{N}$$

$$R = \frac{(0.00213 + 0.00013 + 0.00503)}{N} = \frac{0.00729}{N}$$

Now rate of transfer = 3,000,000 = Mean overall temp. diff. divided by the overall resistance = $\frac{91.0}{0.00729/N}$, whence N = 240 ft. of copper pipe.

Since the velocity of water in the tubes is to be 4 ft. per sec. each tube will handle

$$(4)\quad \frac{(0.625)^2\ (0.785)\ (3600)\ (62.3)}{144} = 1910 \text{ lb. water per hr.}$$

The number of tubes in parallel necessary to give 4 ft. per sec. is 30,000/1910 = about 16 tubes. The length of series travel is 240/16 = 15 ft. This may be divided into two or more passes as desired, but the more the passes, the greater the friction, due to the additional entrance and exit friction losses from the tubes.

OPTIMUM VELOCITY

In the above formulas it will be noted that velocity is an important variable and one largely within the control of the designer. Thus, in air heaters for ventilating systems, driers, humidifiers, etc., one may use a high air velocity, thereby securing a high coefficient of heat transfer, but this is done at the expense of a high pressure drop. If desired, one may go to the other extreme and use a very low air velocity requiring therefore a large heating surface but involving very little pressure drop and therefore only small power consumption. The decision as to the velocity to be employed in the given case requires an economic balance between cost of power and cost of heating surface. The curves in the diagram (Fig. 1) have been calculated for the heating of 10,000 cu. ft. per min. from 70 to 170 deg. fahr., using a steam condensing at 220 deg. fahr. in a heater made of staggered standard 1 in. pipe. The coefficients were figured from the Carrier formula and the pressure drops from the Sturtevant[1] tables. The cost of heating surface was taken as $0.927 per sq. ft. including all installation costs. The combined (overall) efficiency of fan and motor was assumed to be 50 per cent and the cost of power 3 cts. per kw. hr. The drier for which this equipment was designed operates 7200 hr. per year, i. e., 300 days. The charge against the heating surface for depreciation, interest, taxes, maintenance, etc., was assumed to be 15 per cent.

It will be noted that the point of lowest total cost corresponds to an initial air velocity of about 10 ft. per sec. through the minimum clear area. The reason for this is that whereas the charges against heating surface are annual, the power cost necessary to force the air through that heating surface is mounting up every hour of operation.

Note also that the percentage charged off against depreciation and the like is very low. Under conditions where this charge must be higher

[1] B. F. Sturtevant Co., *Data Book for Engineers and Architects*, p. 938, (1917).

even lower velocities are desirable but where power is cheaper the reverse would be true. There is often a tendency on the part of the designer to keep down initial investment by an installation that is in reality inefficient because of excessive operating costs.

SUMMARY

The points to be kept in mind are, first, that mass velocity, viscosity and thermal conductivity are the major factors in determining the coefficient of heat transfer from gases or liquids to solids, and second, that, for accuracy, heat transfers must be calculated by the use of film coefficients, one for each side of the heating surface.

Those interested in further details of this subject are referred to a series of articles by two of the authors which will start in the January issue of the *Journal of Industrial and Engineering Chemistry.*

NOMENCLATURE TABLE

c = Cleanliness and roughness coefficient.

c_p = Specific heat of gas at constant pressure = B.t.u. per lb. gas per deg. fahr. = gm. cal. per gm. gas per deg. cent.

D = Actual inside diameter of pipe in in.

f = A coefficient in simplified Weber equation for air.

h = Film coefficient of heat transfer as B.t.u. per hr. per sq. ft. film area per deg. fahr. drop through the film.

H = Overall coefficient of heat transfer as B.t.u. per hr. per sq. ft. suitable average area of heating surface per deg. fahr. overall temperature difference.

k = Absolute thermal conductivity of fluid as B.t.u. per hr. per sq. ft. per deg. fahr. per ft. thickness. Values of k for solids, liquids, and gases are given in many handbooks. For special liquids, see McAdams and Frost, Journal Ind. Eng. Chem., Vol. 13, No. 1, January 1922.

M = Average molecular weight of fluid.

s = Specific gravity of fluid at temperature and pressure in question in lb. per cu. ft. divided by 62.3.

S = Shape factor. = Area of heating surface in sq. ft. divided by free volume of apparatus in cu. ft. Note that S equals the reciprocal of hydraulic radius taken in ft.

T = Average temperature of gas as deg. fahr. absolute = deg. fahr. temp. plus 460.

u = Average linear velocity of fluid as ft. per sec. = cu. ft. per sec. divided by average clear area in sq. ft.

v = Average mass velocity of fluid as lb. fluid per second per sq. ft. of average clear cross sectional area. Note that $v = u$ multiplied by average fluid density as lb. per cu. ft.

w = Mass velocity at the center line of pipes in the same row, for use in Carrier's equation only.

x = Per cent steam by volume = 100 times the partial pressure of steam divided by total pressure of mixture in the same pressure units.

y = A coefficient in the formula for critical velocity.

z = Viscosity of fluid relative to water at 68 deg. fahr. as unity = absolute c.g.s. viscosity of fluid at temperature in question divided by 0.0100. (For more details see foot-notes 1 and 2 on p. 58.) For water, z is a function of the water temperature (t) in deg. fahr., as indicated by the following approximate equation: $\frac{1}{z} = -0.12 + 0.0161\ t$ which holds between 70 and 180 deg. fahr. The exact values are given in Fig. 2.

DISCUSSION

W. H. CARRIER (written). This is a field of investigation that has been urgently needed by the engineering profession for a long time. It has seemed that the data and theory on heat transfer has been sadly neglected in the past. It is a very important subject, not only to heating and ventilating engineers, but to power plant and chemical engineers as well.

The writers of this paper are to be congratulated on the work they have done on this very much needed line of investigation. I further appreciate that in the brief space allotted to this paper, it is impossible for them to give more than an outline of the work that they have done, and I am in hopes that eventually they will present to this Society the complete engineering data and basis from which their calculations have been derived. Presumably this complete presentation awaits a further experimental development, which I understand from the paper is in process.

I am especially interested in the formula derived on page 60, an equation for condensing vapors as a function of the conductivity and viscosity. This simple relationship appears entirely rational, and I believe that the general form will be borne out with considerable accuracy by further tests, which I understand are contemplated.

Comparing the great variation in viscosity of water, as shown in Fig. 2, page 59, it is quite evident why the comparatively low heat transfers are secured at low temperatures, as in water cooling by refrigeration, etc., also why such low transfers are obtained in the cooling of oil. This subject alone could receive years of experimental study to advantage.

The presentation of the comprehensive example of the method of application is also to be commended, as it makes the subject appear more practical in its application than would the mere presentation of the formula.

The question of optimum velocity treated on page 62, and shown graphically under Fig. 1, page 57, is of the importance that has been too little appreciated by engineers. I would call attention, however, to the fact that the authors have overlooked the previous work that has been done upon this subject in this Society. I would refer to the papers of Busey and Carrier, (see Vol. 19, TRANSACTIONS, 1913, p. 141). This covered the same subject as treated briefly by Professor Lewis, giving the results both for heaters and for piping. The same general conclusions were reached as given by Prof. Lewis, except that a somewhat higher resistance was taken for the

heaters, due in part to the use of different centers than in the data used by Prof. Lewis.

The figures used by Mr. Busey and the writer, however, are thoroughly verified by careful tests by the Buffalo Forge Co. and also by Mr. Soule for the American Radiator Co. The general result found by Mr. Busey and myself, by Mr. Soule for the American Radiator Co., and by Prof. Allen of the University of Michigan, was that the heat transfer coefficient was substantially the same when velocities and areas were chosen in which equal resistances were obtained, regardless of the type of heater used.

This inter-dependence of friction and transmission, I believe has not been fully recognized by the authors of this paper, and I would suggest that this be given a consideration in their general formula. I believe it is possibly accounted for, to some extent, by the shape factor, but as I will try to show later, the probability is the shape factor has a more complicated formula, and would possibly best be replaced by a series of special constants, or simple variables, dependent on the type of heater.

The writer's formula, I believe, is in substantial agreement when applied to fluids, with the form of equation used by Prof. Lewis. The general formula for water, on page 101, March, 1922, JOURNAL, seems entirely rational, with the possible exception of the exponents 0.8 for u. I believe that the heat transfer can be best represented by the reciprocal for the formula for resistance, and all of the data obtained by the writer, as well as data obtained by others, which the writer has examined carefully, can be reduced to the general resistance formula for heat transmission, in which one term is independent of the velocity, but dependent upon the type of surface and the physical characteristics of the liquid, or gas, such as specific heat, density, conductivity and probably viscosity. The second factor of resistance varies as the exact inverse of the velocity, but has a constant factor which includes, and is inversely proportional to the density and specific heat, and the reciprocal of the viscosity. It probably also includes a shape factor, which, in the case of tubular surfaces, is a function of the hydraulic radius, as indicated in Prof. Lewis' paper. The same general type of formula, I believe, is adaptable both to liquids and gases, with slight modifications for gases, due to change of density, with a rise of absolute temperature. I realize that the reciprocal resistance formula can be closely approximated within certain ranges by the exponential formula for velocity. For practical work, one may be used to closely approximate the other, but this exponent is not constant for different forms of surfaces, as has been assumed in Prof. Lewis' paper, and varies from about 0.67 for fluids outside and at right angles to staggered pipes, instead of 0.8, as given by Prof. Lewis, up to 0.8 or slightly greater for flow through tubes.

On this account, I believe it is better to use the reciprocal resistance formula, which is the rational formula for heat transfer. As I will show later, it has a rational basis, and holds exactly for great ranges of velocities, which the exponential formula will not do. It is also much easier to use practically than the exponential formula, which would commend it to engineers if it were even only approximate. I will speak

further regarding this in reference to the tests on heat transmission with air, with which I am more familiar and regarding which there is a large quantity of accurate data available.

WEBER'S FORMULA FOR HEAT TRANSFER WITH GASES

In view of the large quantity of carefully analyzed and consistent data obtained by the writer and others, on the heat transmission from surfaces to air, the writer is forced to disagree quite radically with the relationships expressed by Weber's formula, as given in Prof. Lewis'

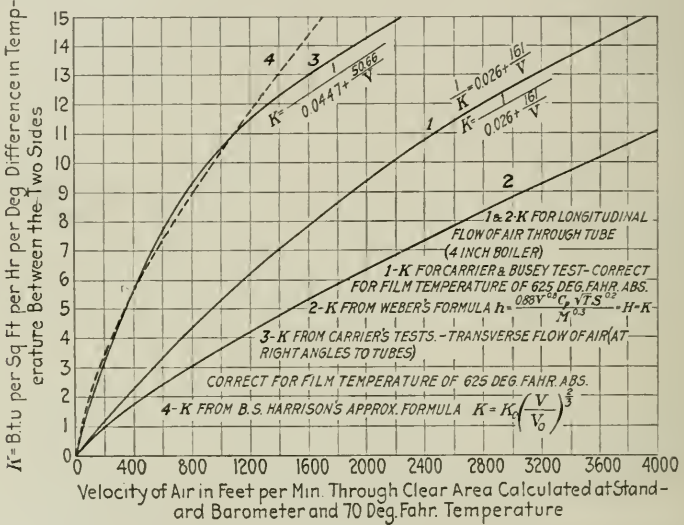
K=Btu per Sq Ft per Hr per Deg Difference in Temperature Between the Two Sides
Velocity of Air in Feet per Min. Through Clear Area Calculated at Standard Barometer and 70 Deg. Fahr. Temperature
1 & 2-K FOR LONGITUDINAL FLOW OF AIR THROUGH TUBE (4 INCH BOILER)
1-K FOR CARRIER & BUSEY TEST- CORRECT FOR FILM TEMPERATURE OF 625 DEG. FAHR. ABS.
2-K FROM WEBER'S FORMULA
3-K FROM CARRIER'S TESTS. - TRANSVERSE FLOW OF AIR (AT RIGHT ANGLES TO TUBES)
CORRECT FOR FILM TEMPERATURE OF 625 DEG. FAHR. ABS.
4-K FROM B.S. HARRISON'S APPROX. FORMULA

FIG. 3. CURVES SHOWING HEAT TRANSMISSION AT VARIOUS VELOCITIES WITH DIFFERENT FORMULÆ

paper. I wish to refer you first, to the paper presented by Frank L. Busey and the writer, on Air Conditioning Apparatus (see Vol. 33, Trans. A. S. M. E., 1911, p. 1055), which includes a study of the theory of convection with forced circulation, with considerable data on the rate at which heat is transferred in various sections of heaters with various air velocities, and constant steam temperature of 227 deg. This same data is more thoroughly analyzed and discussed in the paper on Heat Transmission with Indirect Radiation, which was presented by Frank L. Busey (see Vol. 18, TRANSACTIONS, 1912, pp. 172 to 204). This paper includes complete log of the tests on standard coil pipe heater at 227 deg. temperature, and also a similar test at 50 lb. steam pressure. The data on the latter had not been worked up at the time the first paper was presented before the A. S. M. E., but afterwards, when worked up for the later paper, read before the A. S. H. & V. E., it was found to agree exactly with the formulae derived from the first test, although the heat transmission per degree difference of temperature in the second case was considerably lower than in the first case, a result which is also in

accord with the general formula first deduced. In the second case, the air was heated to a considerably higher temperature, as will be indicated by the log of the test, than in the first case. Nevertheless, the value of K decreases and not increases, as Weber's formula would indicate.

A further refutation of the relationship, expressing the transfer as proportional to the square root of the temperature, is contained in a study of the relative heat transfer in the successive sections of the heater, of which there are eight. I will refer you to the table on page 181, Vol. 18, 1912, TRANSACTIONS A. S. H. & V. E., column 7. On the assumption that there were equal transfer per degree difference, then the ratio of leaving to entering temperature would be constant for each section, as will be seen by the mathematical analysis of the usual transfer formula.

According to Weber's formula, the last section should have a transfer per degree difference proportional to the square root of the ratios of $\frac{612}{492}$ or approximately 10 per cent greater transmission. This is absolutely contrary to fact, as may be seen by the examination of these ratios. The ratio of final or initial temperature difference increases from 0.885 to 0.889, or the relative temperature rise decreases from 0.115 to 0.111, which indicates that the effect of the rise in absolute temperature is just the reverse of that indicated by Weber's formula, and is very much less in effect. This is satisfactorily accounted for in this paper by the rise in the absolute film temperature θ, in which it is shown that R, the factor of constant resistance independent of velocity, varies directly as the absolute film temperature.

In other words, as the density in the film decreases, due to rise of temperature, the resistance to the transfer of heat into the film increases in inverse proportion. The same laws seem to hold true at 50 lb. steam pressure (Fig. 10, p. 189, Vol. 18, 1912, TRANSACTIONS), that for the same film temperature, the same resistance to heat transfer obtained as with 5 lb. steam pressure (compare Fig. 5, page 183). It will be seen that these two reciprocal curves are identical, although one was obtained with 5 lb. steam pressure and the other with 50 lb. steam pressure, corrections in both cases being made for the same film temperature. All of this is quite contrary to the assumptions of Weber's formula, with reference to absolute temperature, as affecting the heat transfer.

The same analysis was made for two sets of tests made at different times by the American Radiator Co. on Vento coils, and also upon pipe coils, both in the latter tests, but one steam temperature was used and condensation was not measured separately in the successive sections, so that the data in this respect are not so conclusive as that here presented.

There is absolutely no getting away from the fact that Weber's formula does not truly represent experimental data, even approximately, in this respect.

Now with reference to the velocity term $v^{0.8}$ this is in fairly good agreement with the experiments between limited ranges, for air or gas, *blowing*

through tubes. It is not in agreement with experimental data, however, for air blowing transversely across tubes. The exponent for this is more nearly 0.67. However, the general form of this equation with respect to v is not rational inasmuch as it does not agree with facts. The facts are that in every case of heat transfer, the rate of transfer per degree difference of temperature approaches a finite value, at infinite velocity. At high velocities, this increases very little with increase of velocity, although it approaches this maximum very rapidly at the lower velocities. This value of theoretical maximum transfer is very much less, especially with gases, than would be indicated by the resistance of the wall or the condensing film. In fact it is only about 1/50 of what would be indicated by the maximum transmission, presumably due to the resistance of the condensing film and the tube wall, so that there is

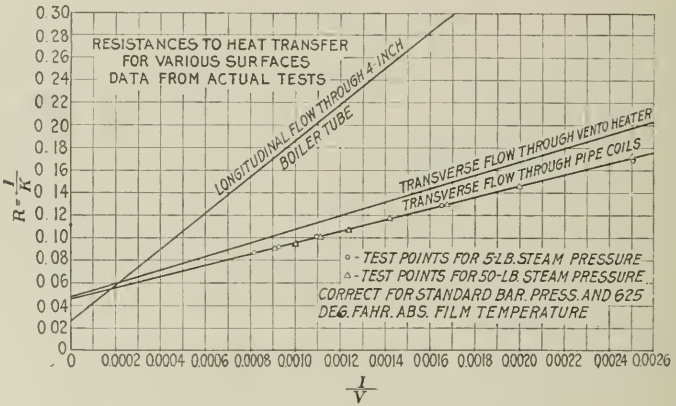

FIG. 4. CURVES SHOWING RESISTANCES TO HEAT TRANSFER FOR VARIOUS SURFACES

evidently a fixed resistance due to the air film which is many times that of the other factors of resistance, and cannot be confused with them.

From the further fact that there is a limiting factor as shown by experiment, to the maximum transfer of heat to the tube surface, the exponential formula cannot be held, as this increases infinitely with infinite increase of velocity, and does not approach a fixed value. The only form of equation which will permit this to approach a fixed value is the reciprocal form of equation, including a constant resistance and a variable resistance, decreasing as the velocity increases. Moreover, the resistance form of equation, that is, the reciprocal equation is expressed by

$$h = \frac{1}{R + B/v}$$

which is more easily used by the engineer in calculations, as well as being the rational form applicable to wide ranges of velocity.

This same equation has been applied to experimental data obtained by Mr. Busey and the writer, and published in the Handbook of Fan

Engineering, by the Buffalo Forge Co., pages 405 and 406[1]. The experimental work on this was conducted on a 4 in. boiler tube, jacketed with steam at 5 lb. pressure. The total air passing through the tube was metered accurately with a calibrated orifice meter, and the heat transfer calculated by the temperature rise. Ranges and average velocities were obtained experimentally from 500 to 4000. I have compared, and am presenting a copy of this curve, Fig. 3, as compared with the corresponding values as given by Weber's formula, on page 58 in Prof. Lewis' paper. This is taken for the average absolute temperature of 555 deg., although it was somewhat higher than this on the low velocities, and somewhat lower at the higher velocities. However, as previously shown, the heat transfer does not vary as a direct function of the absolute temperature.

These curves plotted for the same velocities and transformed to the velocity in ft. per min., instead of ft. per sec., give values approximately two-thirds of those determined experimentally by Mr. Busey and the writer. We think there is either some typographical error, or else the coefficient given is about two-thirds of what it should be. The Weber formula used contains a slight correction in that appearing in the paper, and is as follows:

$$h = \frac{0.88\, v^{0.8} C_p \sqrt{T}\, S^{0.2}}{M^{0.3}}$$

h is taken as the film coefficient of heat transfer, as B.t.u. per hr. per sq. ft. film area per deg. fahr. drop. v is in lb. of air per sq. ft. per sec.

$S = \frac{4}{0.31} = 13$, the internal diameter of the tube being .31 ft.

It will be found that the resistance of the condensing film and the tube wall combined is 0.000685, which gives a conductivity of the condensing film and wall combined as 1460. From this it will be seen that the resistance of the condensing film and wall affects the total resistance in any case only in about the fourth place, so that $H = h$ for air under these conditions. Corrections to be made for the total conductivity as

---

[1]NOTE: In the formula referred to on pages 405 and 406 of Fan Engineering, a correction should be noted. The values given by the curves are correct, and plotted for the experimental data. In the formula given on page 405, there is a mathematical error to which attention should be called, in the calculations of the slope of the resistance line. The correct formula for the resistance, that is $\frac{1}{K}$, is

$$\frac{1}{K} = .026 + \frac{161}{v}$$

and the correct formula for K with air flowing through tubes is

$$K = \frac{1}{.026 + \frac{161}{v}}$$

compared to the film conductivity are so slight that they would not appear in plotting.

Further tests have been made in the use of this reciprocal formula recently, in the experiments conducted on the kata thermometer, and it was found with the three different types of kata thermometers, under entirely different conditions, the losses of convection were completely covered by the reciprocal formula. In fact, so close was this agreement

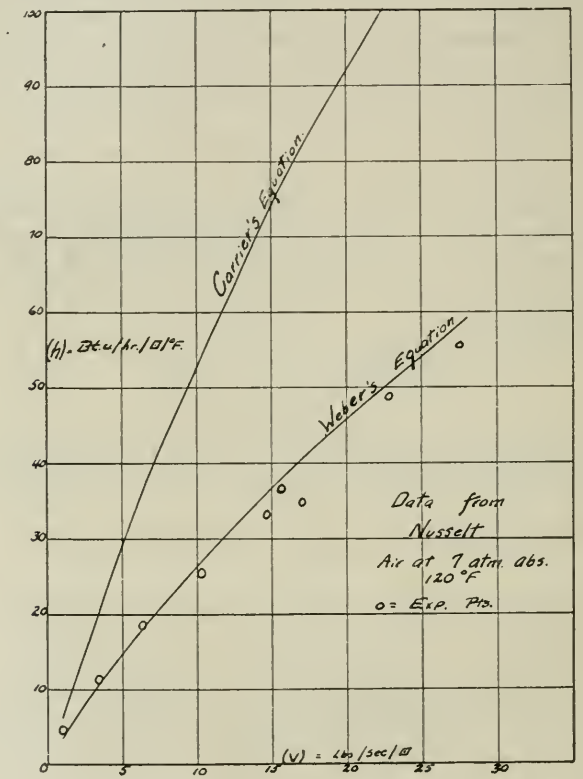

FIG. 5. CURVES SHOWING COMPARISON OF CARRIER'S AND WEBER'S EQUATION WITH NUSSELT'S EXPERIMENTAL DATA TO DETERMINE HEAT TRANSMISSION

that the first results worked up for velocities between zero and 80 feet a minute permitted the determination of equations which were found to hold within 2 per cent on later tests, with velocities ranging from 100 to 2000 ft. per min., which is about as convincing a justification for the rational form of this formula as could be expected. In fact, up to this time the writer had considered this formula might have a more or less empirical basis. This possibility, of course, still exists, but the remarkable consistency of results obtained with this would make it appear that it has a physical basis, rather than an empirical one.

I am presenting herewith a general formula for gases for all tempera-

tures, pressures and velocities, based on air in a 4 in. tube, which is as follows:

$$K = \frac{C_p}{0.00425 \left(\frac{T_f}{p\,M} + \frac{292}{G\,S^{0.2}} \right)}$$

where T_f = absolute film temperature.

p = absolute pressure in lb. per sq. in.

M = molecular weight of gas.

G = lb. gas per hr. per sq. ft. clear area.

S = surface factor as in Weber's formula.

I am including in this formula Weber's shape factor, and assuming that his exponent 0.2 is approximately correct. I do not see why this should be materially different, however, from the reciprocal of the shape factor used in Prof. Lewis' paper for water, which is 0.17. The advantage of the approximate exponent of 0.2, however, is that it is more easily calculated, being the square root of the cube root, and therefore avoiding the use of logarithms. I believe the formula as here given will apply approximately for gases other than air. It should be noted, however, that the two resistance factors in the denominator are affected by different things. The film factor is affected by the specific heat of the gas, its molecular weight, the pressure and the absolute film temperature. In the second term, if velocity is expressed as mass flow in terms of pounds of gas per hour per square foot of free area, then the only other factors affecting this transfer are the shape factor, which does not affect the first term in any way, and the specific heat, which also affects the first term. Pressure, absolute temperature and molecular weight of the gas are all taken into account in determining the value G and therefore do not appear again.

I am very certain that an equation of this form is much more trustworthy than Weber's formula, for other gases than air, for the very reason that Weber's formula does not check with experimental data on air and therefore could not be expected to be reliable for other gases. I realize that this general formula appears quite complicated, but it is necessary in representing the conditions of heat transfer with exactness. The conductivity as controlled by film resistance, which is given in the first term of the denominator, must be clearly distinguished from the second term, which represents the effect of convection due to velocity alone. (In this connection, I would call attention to the fact that the writer's formula, to which the paper under discussion refers, is not in original form, as given by the writer, but is changed to a form in which the two resistance factors are not clearly distinguished, as in the original, neither is it quite as convenient for calculation. The original form is as follows:

$$K = \frac{1}{a + b/v}$$

For air blowing through steam heated tubes, with film temperature of

approximately 625 deg. absolute and standard barometric pressure, this becomes

$$K = \frac{1}{0.026 + 161/v}$$

v is the velocity in feet per minute of air having a density of 0.075, that is, this is the velocity at 70 deg. at standard barometric pressure. It is in reality the equivalent of the mass velocity.

It will be noted in the writer's general formula as compared with Weber's formula, that the exponent for the velocity is 1.0 instead of 0.8. This is due to the fact that there are two terms in the writer's where there is but one term in Weber's formula. Within limited ranges, both forms can be made to give corresponding results if proper coefficients are chosen. For higher velocities, however, Weber's exponential formula does not hold, as already pointed out.

It will also be noted that in the writer's equation, the exponent for the molecular weight (M) is taken as one, while in Weber's formula, it is taken as 0.3, but applying to the whole equation. In the writer's formula, it affects independently only the first term, and therefore has a smaller effect on the entire value of conductivity, it belongs rationally in this term and should affect this resistance inversely as a first power.

It should be particularly noted that the absolute temperature (T_f) used in the first term, is not the mean absolute temperature of the air, but is the mean film temperature, which is dependent upon the temperature of the conducting wall, upon steam temperature, upon the temperature of the air, and to some extent, upon the velocity. The general formula for the determination of the film temperature, if it is desired to determine it accurately, is given in the original papers above referred to, and was found to be as follows:

$$T_f = T_s - R\ K\ (T_s - T_m)$$

T_f is absolute temp. of surface film; T_s absolute temp. of steam; R is approximately the constant resistance term $= 0.026$ for flow through tubes. K is approximately total rate of heat transmission; T_m is absolute temp. of air. For approximate work, an appropriate value of the film temperature may be assumed with but slight error. It is but slightly lower than the temperature of the transmitting wall, except at very high velocities, or high densities. It should be further remarked that this formula has been applied successfully not only to air at atmospheric pressure, but to transmission with air under several atmospheres pressure. Commercial tests indicated that the formula held with exactness. It is also found to hold with different steam temperatures, as well as different air temperatures through a wide range.

I do not wish to discourage the very valuable work that is being done by Prof. Lewis and his associates along the line of a general formula, but I think it is certainly in the interests of scientific and engineering accuracy that these discrepancies be clearly pointed out with a hope that some common ground may be reached.

APPLICATION OF WEBER'S FORMULA TO 4 IN. BOILER TUBE

Mean Air Temperature—95 deg. fahr.

$$h = 0.88\ v^{0.8}\ \left(\frac{Cp\ \sqrt{T}\ S^{0.2}}{M^{0.3}}\right)$$

4 in. boiler tube
0.134 in. = thickness of wall
Inside diameter = 3.73 in.
Molecular weight air = $M = 28.8$
Specific heated air = $Cp = 0.241$

$$\text{Shape factor} = S = \frac{\pi \times 0.31}{\frac{\pi \times 0.31^2}{4}} = \frac{4}{0.31} = 13$$

Absolute temperature = $T = 460 + 95 = 555$ deg.

$$\frac{Cp\ \sqrt{T}\ S^{0.2}}{M^{0.3}} = \frac{0.241 \times \sqrt{555} \times 1.67}{2.74} = 3.46 = \text{const.} = C$$

Mass velocity ft. per sec. $= v = \frac{V}{60} \times 0.075$ where V = vel. ft. per min.,

or $v = 0.00125\ V$.

$$\text{Resistance steam} = R_s = \frac{z}{2200K} = \frac{z}{725} = \frac{1}{3.8 \times 725} = 0.000363$$

$$\text{Resistance air} = R_a = \frac{1}{h} = \frac{1}{0.88\ v^{0.8}} \times \frac{1}{C} \text{ and } \frac{1}{C} = \frac{1}{3.46} = 0.289$$

$$\text{Resistance tube wall} = R_w = \frac{0.13}{K \text{ iron}} = \frac{0.13}{403} = 0.000322$$

$$\text{Total resistance} = R_z = R_a + (R_s + R_w)$$
$$= R_a + 0.000363 + 0.000322 = R_a + 0.000685$$

$$\text{Transmission} = K_z = \frac{1}{\frac{1}{C\ 0.88\ v^{0.8}} + 0.000685}$$

$$= \frac{1}{\frac{0.289}{0.88\ v^{0.8}} + 0.000685}$$

THE AUTHOR (W. K. Lewis). The linear relationship to which Mr. Carrier refers, obtained by plotting the reciprocal of velocity against the reciprocal of heat transfer, does not hold for low velocities, because at

low velocities the character of the motion changes. The particles flow in viscous rather than turbulent motion, and because of the absence of mixing by eddy currents the heat transfer changes.

This may seem a hypothetical condition not met in practice. However, a few years ago, we investigated trouble encountered in the operation of a shelf drier in which a sheet material was being dried on trays 2 in. apart, placed in a number of compartments in series, the air flowing through these trays at a velocity of about 200 ft. per min. The air was pre-heated before entering the first compartment, but was not further heated until it had passed through several compartments. The sheets started to dry only where the air first touched them, not over one-fourth of the area of the sheet being affected. However, the air leaving the wet surface of one of these sheets would start to dry the sheet in the next compartment, meaning that the air had ceased to dry and yet was not saturated. The explanation is that the film of air next to the sheet was saturated and hence would no longer take up moisture. The air midway between the sheets was unsaturated, however. When all the air was well mixed in flowing from one compartment to the next, unsaturated air was brought into real contact with the sheets and drying started again.

In heating and ventilating work where natural draft is used velocities may fall to such low values as to cause straight line motion of this sort, in which case none of the formulæ for heat transfer referred to apply.

(WRITTEN DISCUSSION): We have read with interest Mr. Carrier's discussion of our article, and in replying thereto have classified his criticisms so as to be able to discuss them separately.

AIR INSIDE PIPES—RELIABILITY OF EQUATION

In his first plot, Mr. Carrier finds that Weber's equation calls for coefficients of heat transfer about 30 per cent lower than the data of Carrier and Busey for air in a 3.73 in. pipe. Mr. Carrier concludes that the Weber equation is incorrect for air inside pipes, and offers a general formula for air as being more reliable than the Weber formula. Fig. 5, 6 and 7 show actual experimental points obtained by several investigators, and in addition, two lines are drawn on each plot for each set of experimental conditions, one calculated from the new Carrier equation, and the other calculated from the Weber equation.

Fig. 5 shows Nusselt's data[1] for air at 7 atmospheres and at an average temperature of 120 deg. fahr. in a 0.868 in. inside diameter pipe for a 28-fold velocity range. In applying the new Carrier equation, the average film temperature was taken as 625 deg. fahr. absolute. The Weber equation is seen to compare well with the experimental points, while the Carrier equation is about 100 per cent higher.

Fig. 6 shows Josse's data[2] for air at 1 atmosphere and 120 deg. fahr. in a 0.905 in. pipe for a 45-fold velocity range. The Weber equation again compares well with the experimental points, while the Carrier equation is about 40 per cent higher.

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[1] Mitteilungen uber Forschungsarbeitungen, (Berlin), Vol. 89, 1910.

[2] Zeitschrift fur Deutsche Ingenieure, 1909.

Fig. 7 shows Robinson's data[1] for air at 1 atmosphere and 130 deg. fahr. in a 1.188 in. pipe over a velocity range of 3-fold. The Weber equation is about 20 per cent higher than experimental points, while the Carrier equation is about 60 per cent higher.

GASES OTHER THAN AIR INSIDE PIPES—RELIABILITY OF EQUATIONS

Mr. Carrier states, "I am very certain that an equation of this form is much more trustworthy than Weber's formula, for gases other than air, for the very reason that Weber's formula does not check with experimental data on air and therefore could not be expected to be reliable

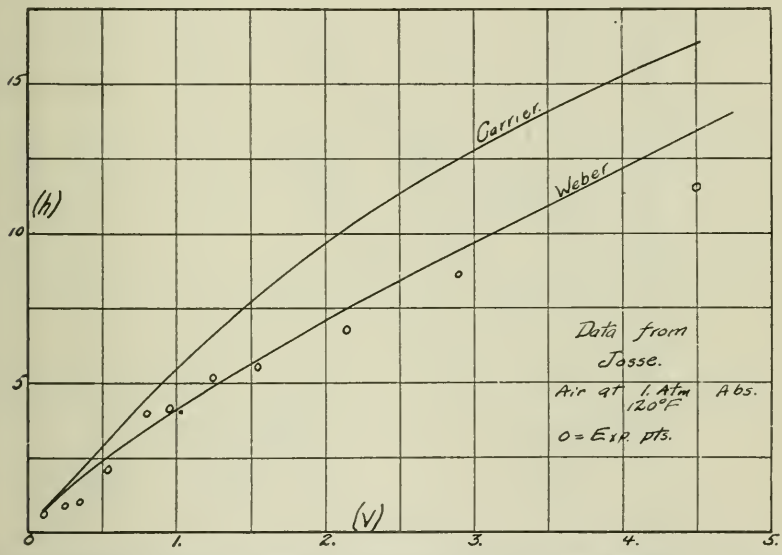

FIG. 6. CURVES SHOWING COMPARISON OF CARRIER'S AND WEBER'S EQUATION WITH JOSSE'S EXPERIMENTAL DATA TO DETERMINE HEAT TRANSMISSION

for other gases." He also states: "I believe that the formula here given will apply approximately for gases other than air."

We have shown above that Weber's equation does check with considerable experimental data for air far better than Carrier's. Fig. 8 shows Nusselt's experimental data for carbon dioxide gas at 1 atmos. and 120 deg. fahr. in 0.868 in. pipe over a 20-fold velocity range. It is seen that Weber's equation checks with the data on carbon dioxide within 10 per cent, while Carrier's equation is 60 per cent higher than the experimental points.

NATURE OF VELOCITY FUNCTION—AIR INSIDE PIPES

Mr. Carrier believes that heat transfer coefficients may be represented more accurately by means of a *reciprocal resistance* formula than by an *exponential* one. Mr. Carrier states: "The facts are that in every case

[1] Private communication from Prof. C. S. Robinson, M.I.T.

of heat transfer, the transfer rate per degree difference approaches a finite value, at infinite velocity." He further states "I realize that the reciprocal resistance formula can be closely approximated within certain ranges by the exponential formula for velocity." He adds: "As I will show later, it (the reciprocal resistance formula) has a rational basis, and holds for great ranges of velocities, which the exponential formula will not do."

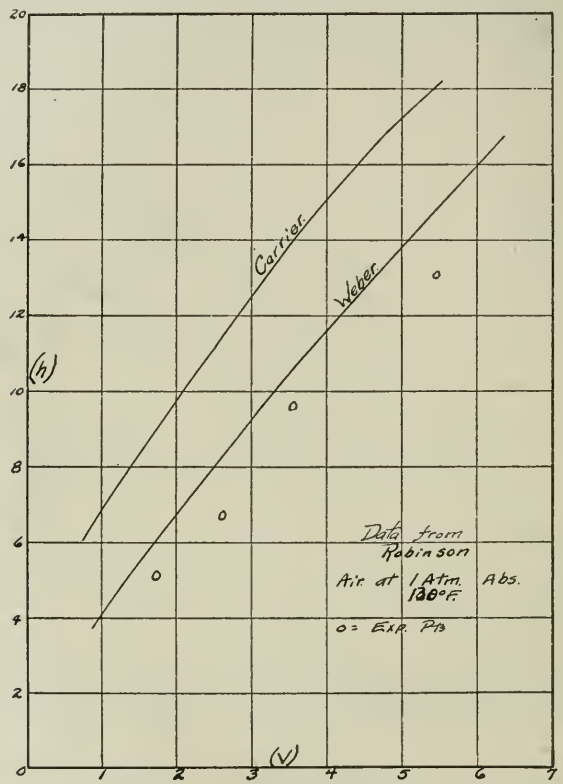

FIG. 7. CURVES SHOWING COMPARISON OF CARRIER'S AND WEBER'S EQUATION WITH ROBINSON'S EXPERIMENTAL DATA TO DETERMINE HEAT TRANSMISSION

We will examine Mr. Carrier's statements by comparing the two methods of plotting under discussion with Nusselt's data for *air,* as this data extends over a large range of velocities. In these plots h represents the coefficient as B.t.u. per hr. per sq. ft. per deg. fahr and v represents mass velocity as lb. air per sec. per sq. ft. of cross section. If Mr. Carrier is correct, $1/h$ plotted versus $1/v$ will give a straight line "for great ranges of velocity," while a plot of h versus v on logarithmic paper *will not* give a straight line over a great range of velocity.

Fig. 9 shows that for Nusselt's data on air, h is an exponential function of v over a 28-fold range, the exponent of velocity being about

0.76. Fig. 10 shows the same data plotted as reciprocals, and it is seen that the line is *not* straight, even taking the velocity range as only 9-*fold*. If the range is extended to 28-*fold*, the curvature is accentuated. (Similar plots could be made using the other data, but in the case of Josse's data, the points deviate considerably from one another, and Robinson's data extends over such a short range that both methods are satisfactory).

These plots show just a reverse of what Mr. Carrier stated. While we do not claim that the exponential function is rigidly accurate, these plots show that over a great range of velocity the hyperbolic is a much poorer approximation than the exponential function for the film coefficient of heat transfer.

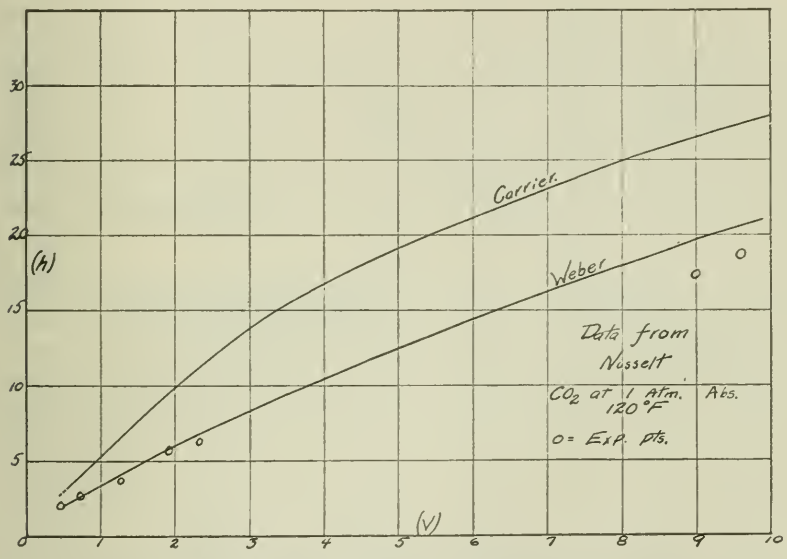

FIG. 8. CURVES SHOWING COMPARISON OF CARRIER'S AND WEBER'S EQUATION WITH NUSSELT'S EXPERIMENTAL DATA ON CO_2

GASES INSIDE PIPES—EFFECT OF TEMPERATURE

Mr. Carrier's formula for air *inside* pipes calls for a decrease in the value of the coefficient of heat transfer as temperature goes up. The best data available on this question of influence of temperature seem to be that of Fessenden and Haney[1] and Babcock & Wilcox[2]. Both of these investigators find a very marked increase in the coefficient with increase in temperature, as shown in Fig. 11.

The Babcock & Wilcox data indicate an increase roughly proportional to the one-third power of the absolute temperature while the Fessenden data follows approximately the two-thirds power. The exponent of 0.5 was chosen by Weber on the basis of theoretical kinetic considerations. It is obvious that the data as to temperature effect is not precise, but

[1] Heat Transmission Through Boiler Tubes, Univ. of Mo. Bulletin, Vol. 17, No. 26, 1916.
[2] Experiments on the Rate of Heat Transfer from a Hot Gas to a Cooler Metallic Surface. The Babcock & Wilcox Co., 1916.

inasmuch as the exponent of 0.5 lay between the values of the two investigators quoted, Weber retained it.

It will be noticed under the first sub-heading of this discussion that at low temperatures the formula of Mr. Carrier is from 40 to 100 per cent high. As the temperature goes up his formula indicates decreasing values of h, while the Weber formula indicates increasing values. Obviously at some temperature the Carrier formula will give results identical with that of Weber but at still higher temperatures its results will be too low. As luck will have it the Weber and Carrier formulas become identical just below the lower end of the experimental temperature range of these experiments, and for the low temperature points the error of the

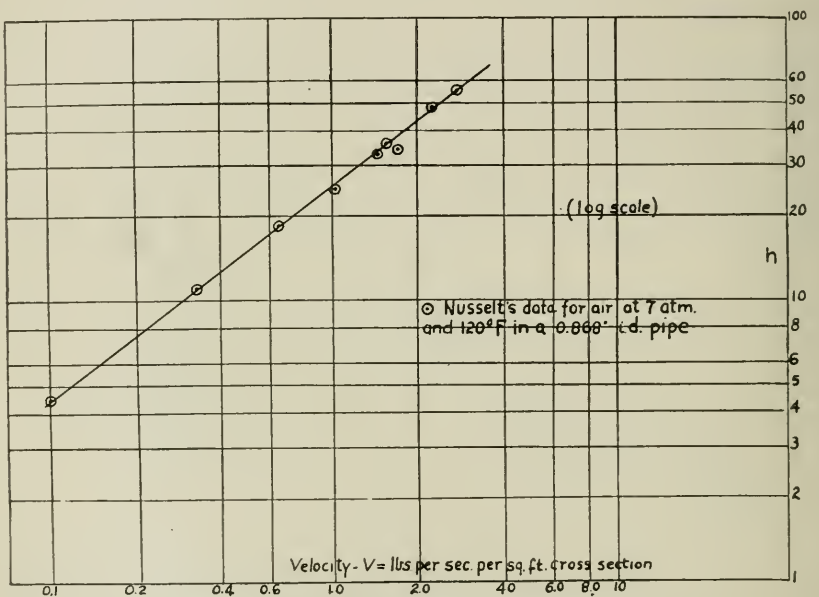

FIG. 9. LOGARITHMETIC PLOT OF NUSSELT'S DATA, THE STRAIGHT LINE SHOWING THAT H IS AN EXPONENTIAL FUNCTION OF V

Carrier formula is only about 25 per cent. At higher temperature Carrier's equation is much lower than the data of either Babcock & Wilcox or Fessenden, whereas Weber's equation checks the latter.

The authors feel that the weakness of Mr. Carrier's formula for the coefficient of heat transmission from gases inside pipes lies in the small amount of data upon which it is based and the fact that his data represents a single set of experiments. We feel that a formula, to be of real value in the problems of engineering design, must be based upon extensive data collected by a large number of investigators operating under varying conditions. It is obvious that from time to time epoch-making investigations will appear which will require the discarding of previous data. On the other hand, we can see no evidence of such epochal character in Mr. Carrier's work and we feel that with the preponderance of the evidence

so strongly against him we cannot disregard the data of such able investigators of world-wide reputation as Nusselt, Josse, Fessenden and Haney, and others quoted above.

There is available in the literature considerable data on heat transfer other than that referred to above. Much of this was used by Weber in deriving his coefficients and checks his formula. So far as is known to us the only data which deviates widely from Weber, either has obvious sources of serious error or else is open to question because of lack of description of experimental method.

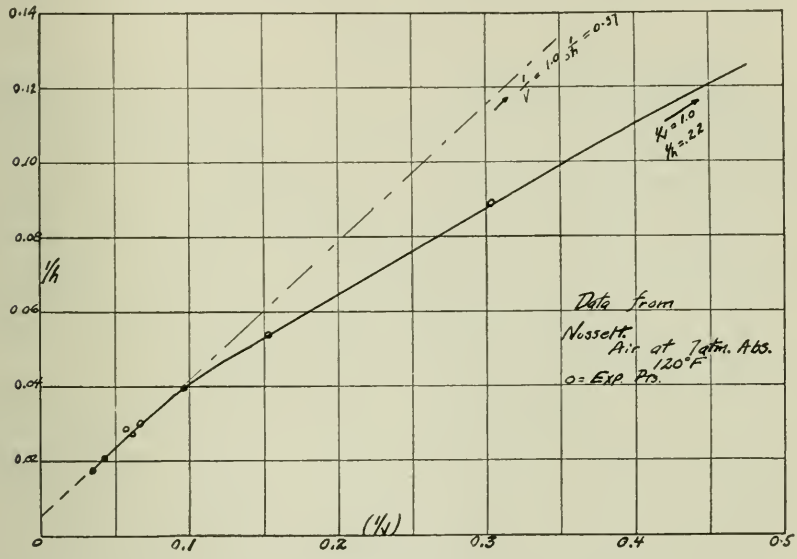

FIG. 10. THE SAME DATA SHOWN IN FIG. 9 PLOTTED AS RECIPROCALS

GASES FLOWING OUTSIDE AND AT RIGHT ANGLES TO PIPES

In our paper, we stated that the Weber equation was derived for gases *inside pipes.*

For *air outside pipes* (see page 57) we quoted Carrier's formula. For air outside pipes, we use Carrier's formula as stated in our article (see page 62, under *Optimum Velocity*). On page 58 we stated "for gases *outside* and at right angles to staggered pipes, the writers employ a modified form of Weber's equation." This should read, "for gases other than air, etc., the writers employ a modified form of Weber's equation." We further stated "this has been tested only for the flow of air." This last was apparently misleading. This modified form was obtained as follows: Data for air outside staggered pipes was compared with what the Weber equation for air *inside* pipes would give for the same velocity, temperature and hydraulic radius, and the ratio so obtained was introduced into the Weber equation for *gases inside pipes,* thus giving a formula *for gases*

other than air outside pipes which could be used as a rough approximation in the absence of experimental data. We regret that the above details were not incorporated in our article.

Mr. Carrier seems quite disturbed because the modified Weber equation for gases outside staggered pipes calls for a small increase in the coefficient with rise in temperature, whereas data of Carrier and Busey show a very slight decrease with a small rise in temperature. Personally we do not take such a small effect so seriously. It has been shown in Fig. 11 that there is a very real increase in the coefficient with increase in

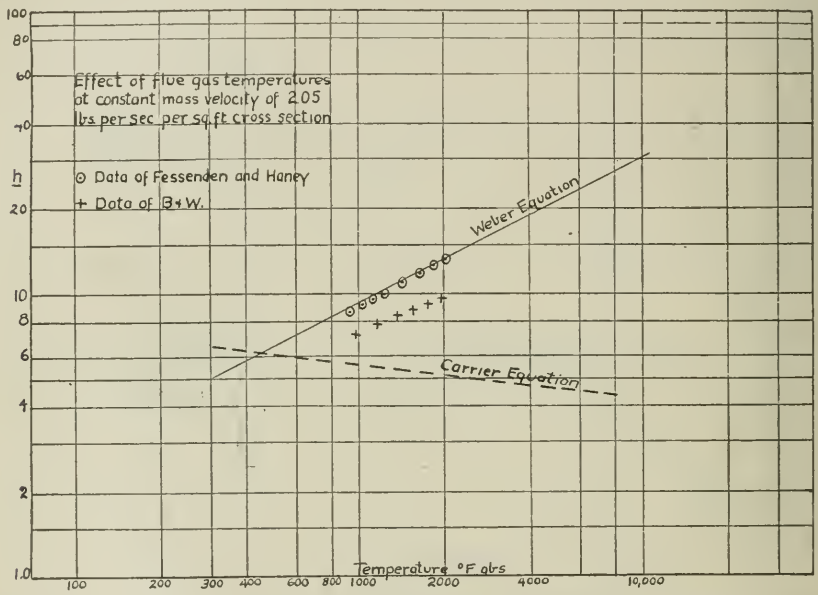

FIG. 11. CURVES SHOWING COMPARISON OF THE WEBER AND CARRIER EQUATIONS WITH EXPERIMENTAL DATA

temperature for the flow of gases *inside pipes*. It will be noted that in the data of Fig. 11 the range is 1,000 deg. rather than 200 deg. fahr. in Mr. Carrier's experiments. It seems incredible to us that the influence of temperature on heat transfer can actually be opposite in effect when the air is outside staggered pipes than what it is when inside. It is obvious that the available data for air flowing outside and at right angles to staggered pipes extend over an insufficient range to answer the question experimentally.

INTERDEPENDENCE OF HEAT TRANSMISSION AND FRICTION

With reference to the *interdependence of heat transmission and friction,* an article from this laboratory discussing this subject as applied to gases and liquids flowing *inside* pipes is soon to appear in the *Journal of Industrial and Engineering Chemistry.*

No. 619

A STUDY IN HEAT TRANSMISSION WITH SPECIAL REFERENCE TO BUILDING MATERIALS

By F. C. Houghten, Pittsburgh, Pa.
Member

THE general question of heat transmission is one which concerns the mechanical engineer perhaps more than any other one subject, and is at the same time one of the most illusive problems to solve. This same question presents itself in many different ways in various fields of engineering. The transmission of heat through boiler tubes is one which has been grappled with by a great many of the best engineers of the world, and though constants sufficiently accurate for most work have been obtained, the general laws are not yet sufficiently established to command the confidence of those interested in the subject.

The heat transmission of refractories used in furnace construction and insulation has also taxed the best skill of the research engineer. Heat insulators for covering steam and hot water piping is of such importance that a great deal of time and money has been spent in research work on this phase of the work alone, resulting in constants sufficiently accurate for the average engineering job, and yet, the research engineer working on the problem admits that very little is really known about the fundamental laws governing the flow of heat through and from the surface of such materials. A great deal of work has also been done on such insulators as are used in refrigeration with the establishment of workable constants. The United States Bureau of Standards has done considerable along this line and has published in the October, 1920, issue of the Journal, p. 625, a series of constants which are no doubt very reliable.

The question of heat transmission through building materials is one which vitally concerns the heating and ventilating engineer, especially the heating contractor. In fact the basis of all calculations on which contracts for heating are based, is the heat transmission constants for the material entering into the construction of the building to be heated, and yet, those of us associated with the Research Laboratory hear frequent confessions from heating engineers, of utter lack of confidence in any of the constants to be had. The lack of consistency among constants used by heating engineers is well illustrated by the results of a questionnaire sent to twenty-five heating engineers by Mr. Arthur K. Ohmes (see Transactions, A. S. H. & V. E., Vol. 22, p. 487). This questionnaire requested an estimate of the amount of heating required for a simple suite of rooms

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January 1922.

for which full specifications were given. The eight estimates offered varied by over 50 per cent.

While a great deal of good and reliable work has been done on the determination of these constants, up until the establishment of the Research Laboratory, the most of this work was done by colleges and universities without the assistance of outside capital. While this work is to be commended and encouraged in every way, and while an immeasurable amount of good has been derived from this work, it has not brought forth the kind of results arrived at by a laboratory backed by the necessary funds.

The transfer of heat from a warm room through a simple wall and out into the cold air is a complicated phenomenon. First the heat is transmitted to the surface of the wall by radiation from the warm objects in the room, and by convection currents of the air. After passing through the warm surface, the heat is transmitted through the material to the cold surface of the wall by conduction. Heat is lost from the cold surface of the wall by radiation and convection to the surrounding cold objects and to the cold outside air.

SYMBOLS AND DEFINITIONS

H = Total heat flow per hour through the wall per sq. ft. of surface for actual temperature difference.
t_w = Temperature of air on warm side of wall.
t_c = Temperature of air on cold side of wall.
t_1 = Temperature of warm surface of wall.
t_2 = Temperature of cold surface of wall.
c = Thermal conductivity constant for any homogenous material in the wall. B.t.u. per hr. per sq. ft. per deg. temperature drop per in. thickness.
C = Thermal conductance of complete wall in B.t.u. per sq. ft. per hr. per deg. temperature drop per actual thickness.
k_w = Surface transfer constant for warm side of wall. B.t.u. per sq. ft. per hr. per deg. temperature drop between air and surface of wall.
k_c = Surface transfer constant for cold side of wall. B.t.u. per sq. ft. per hr. per deg. temperature between cold surface of wall and air.
U = Total transmission constant for the complete wall. B.t.u. per sq. ft. per hr. per deg. temperature drop between the air on the warm side of the wall and the air on the cold side of the wall.
H_r = Heat transmitted to the surface of the wall by radiation. B.t.u. per sq. ft. per hr.
H_n = Heat transmitted to the surface of the wall by convection. B.t.u. per sq. ft. per hr.
d = Thickness of the wall.

THEORY OF HEAT FLOW

Consider the simple wall of thickness d, Fig. 1. Let t_w and t_c represent the warm and cold air temperature inside and outside the room, sufficiently far from the walls to be unaffected by them. Let t_1 and t_2 represent the temperatures of the warm and cold surfaces of the wall. Heat is given off by all objects in the room, including the molecules of the air, in accordance with the Steffans and Boltsman law. The amount of heat given off by every surface in the room by radiation is equal to

$$H = RAT^4 \qquad (1)$$

where R is the radiation constant for each particular surface, A the area,

and T the absolute temperature. Heat is given off in accordance with this law from all objects in the room, including the surface of the walls. The objects in the room give heat to each other and to the walls, and the various parts of the walls give heat to each other and to the objects in the room. The net heat loss per hour from the objects in the room to each sq. ft. of wall by radiation is

$$H_r = R(T_w^4 - T_1^4) \qquad (2)$$

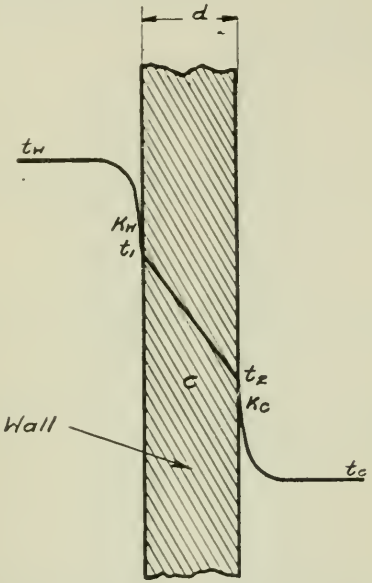

FIG. 1. TEMPERATURE GRADIENTS THROUGH A SIMPLE WALL.

Heat is transferred from the warm air of the room to the cooler surface of the walls by convection currents in accordance with the following law:

$$H_n = NV^{\frac{1}{n}}(t_w - t_1) \qquad (3)$$

Where, H_n is the heat transmitted to each sq. ft. of the wall per hour. N is a constant which depends upon the character of the surface, V the velocity of the convection currents, and $(t_w - t_1)$ the difference in temperature between the air and the wall.

The total heat lost to each sq. ft. of wall surface per hour is therefore,

$$H = H_r + H_n = R(T_w^4 - T_1^4) + NV^{\frac{1}{n}}(t_w - t_1) \qquad (4)$$

It is extremely difficult, if not impossible, to evaluate R, N, n, and V. For this reason the total heat absorbed by each sq. ft. of the wall surface per hour is expressed by H in the following equation:

$$H = k_w(t_w - t_1) \qquad (5)$$

The surface transmission constant is called k_w and must be determined

experimentally for each surface. This is an empirical expression and is not strictly true since it takes no cognizance of the temperature of the objects in the room which temperature may greatly effect the heat transmitted to the walls by radiation. For instance, in a room whose air is at 70 deg. fahr., but which contains objects at a much higher temperature, there will be more heat transmitted to the walls by radiation than in a room in which all the objects are also at 70 deg. fahr. This accounts for the high value of k_w found in some so-called hot boxes in which the source of heat is at a very high temperature as, for instance, a bank of lamps.

The transmission of heat from the outer surfaces of the wall is expressed by the following equation,

$$H = k_c\ (t_2 - t_c) \tag{6}$$

This equation is similar to equation (5) and is for the same reason only an approximation.

The flow of heat through the material of the wall itself is given by the equation

$$H = \frac{A}{d}\ c\ (t_1 - t_2) \tag{7}$$

where A is the area, d the thickness, and $(t_1 - t_2)$ the temperature drop between the two surfaces of the wall, and c a constant which is called the conductivity of the material. In this equation $\frac{Ac}{d}$ expresses the conducting value, or thermal conductance, of the particular part of the material under consideration. The inverse of this expression $\frac{d}{Ac}$ may be called the thermal resistance the material offers to the flow of heat, and is analogous to the expression of electrical resistance. Equation (7) then becomes

$$H = \frac{t_1 - t_2}{\frac{d}{Ac}} = \frac{\textit{temperature drop}}{\textit{thermal resistance}} \tag{8}$$

which corresponds to the equation,

$$I = \frac{E_1 - E_2}{R} = \frac{E}{R} = \frac{\textit{potential drop}}{\textit{electrical resistance}} \tag{9}$$

expressing Ohm's law. The expression $(t_1 - t_2)$ which is the temperature drop corresponds to the e.m.f. or the voltage drop in the electrical analogy. The determination of the heat resistance or conductivity then becomes similar to the determination of electrical conductivity.

While the flow of heat through a solid is very similar to the flow of electricity, there are several differences which add to the difficulty of the determination of the heat constant.

1. The flow of electricity along a conductor is a phenomenon of the electrons of which the atoms of the molecule are composed. The flow of heat through non-metallic substances is a function of the molecules alone.

2. While there is no such thing as a perfect electric insulator or conductor, it is very easy to get materials which for all practical purposes are perfect insulators. Hence, it is easy for practical purposes to confine the flow of electricity to the particular conductor under consideration, while it is next to impossible to confine the flow of heat to any given path. Herein lies the greatest difficulty in the study of heat transmission.

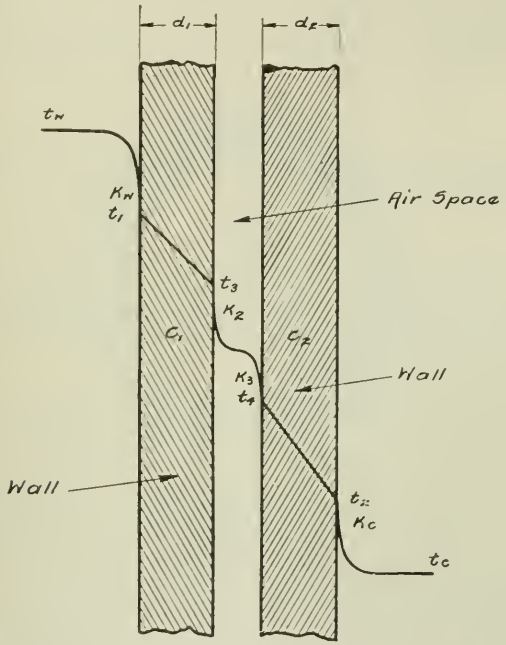

FIG. 2. TEMPERATURE GRADIENTS THROUGH A COMPOUND WALL.

3. It is very easy to measure electrical energy. In fact, the flow of electricity can be measured with an accuracy of 0.001 per cent by the magnetic effect of the passing current, while there has been no such phenomenon found, by which heat in motion may be detected.

Equations (5), (7), (6), expressing the heat transmitted from the warm air to the wall, through the wall, and from the outer surface of the wall to the cold outside air, are of no interest to the heating engineer, excepting as they are means to the end of determining the total transmission constants, expressing the heat transfer from the warm air of the room to the cold outside air.

The total heat loss per hour from the warm air inside the room to the cold air outside is given by the following equation,

$$H = UA\ (t_w - t_c) \qquad (10)$$

where H is the heat passing through the area A of the wall. t_w and t_c are the inside and outside air temperatures and U the total transmission constant.

While U may be determined experimentally, and is often so determined for simple walls by the box method, it is usually more satisfactory to determine U for complicated walls from the surface transmission and conductivity constants k_w, k_c, c as follows,

$$\frac{1}{U} = \frac{1}{k_w} + \frac{1}{k_c} + \frac{d}{c} \tag{11}$$

For the compound wall Fig. 2

$$\frac{1}{U} = \frac{1}{k_w} + \frac{1}{k_c} + \frac{1}{k_1} + \frac{1}{k_2} + \frac{d_1}{c_1} + \frac{d_2}{c_2} \tag{12}$$

where d_1, d_2, etc., are the thicknesses of the component parts of the wall having conductivities c_1, c_2, etc.

METHODS OF DETERMINING HEAT TRANSMISSION

The many investigations of heat transmission through insulating and building materials conducted in the past, may be classified in six groups, depending upon the method used.

A.—Ice box method	D.—Hot air box method
B.—Oil box method	E.—Hot plate method
C.—Cold air box method	F.—Guarded box method

A. The ice box method. In investigations using this method, a box made of the material to be tested and lined with sheet metal, is filled with ice and placed in still air of constant temperature. The temperature of the outside air and the rate at which the ice melts are the necessary data for determining the constants.

B. The oil box method. A box made of the material to be tested, and lined on the inside with sheet metal, is filled with oil which is heated to the desired temperature electrically and stirred. The temperature of the oil, the temperature of the outside air, and the electrical energy used in heating the oil are the necessary data for determining the constants.

C. The cold air box method. The air in a box made of the material to be tested is kept at a lower temperature than the outside air by suspending a box of ice in the interior. In some cases the box of ice has been replaced by refrigerating coils. The temperature of the air inside and outside the box, and the rate at which the ice melts are the data necessary for determining the constants.

D. Hot air box method. Various modifications of this method have been used in more investigations than any other. A box made of the material to be tested contains air which is maintained at the desired temperature by means of electrical heaters. In some cases an electrical fan is used to circulate the air. The temperature of the air inside and outside

the box and the electrical input give the data necessary for determining the total transmission (air to air) constants. If temperatures of the inner and outer surfaces of the material are also observed, conductivity and surface transmission constants may be determined at the same time.

The chief advantage claimed for the hot box method is that it gives values for total transmission from air to air, as well as, conductivity and, both inside and outside, surface transmission constants. In most work done with this type of box, however, the value of k_w is much higher, in instances as much as four times higher, than the value of k_c. While it has not been established that this is not to be expected it is generally supposed that the two values should not show much variation. In fact most experimenters using the hot box method discard the value of k_w and replace it as being equal to k_c. (Bulletin 102 Engineering Experi-

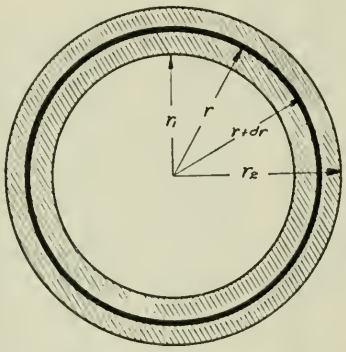

FIG. 3. CROSS-SECTION OF SPHERICAL BOX FOR DETERMINATIONS OF HEAT TRANSMISSION.

ment Station, University of Illinois. Also Bulletin 30 Engineering Experiment Station, Pennsylvania State College.) This increased value for k_w is probably accountable to either one or both of two causes. *First,* excessive direct radiation from a heating element of rather high temperature, in many cases incandescent lamp filaments. *Second,* to taking inside temperature readings too near the surface of the material.

Disadvantages of the hot box method are: (1) Unequal temperatures at the top and bottom of the box, if no fan is used, resulting in unequal heat flow through the six sides. (2) If the air is stirred with a fan, as in many cases, the data do not apply to still air conditions and it is difficult, if not impossible, to determine the air velocity over the inside surfaces. (3) It is difficult to construct a box out of many kinds of material. (4) It is hard to determine the equivalent cross sectional area of the material through which the heat passes, that is, to correct for so-called corner and edge losses. For most work the mean between the inside and the outside surfaces of the box has been assumed to be the equivalent cross sectional area. While this seems to give good results and is generally accepted, it is not quite true. The limits of error resulting from this assumption are hard to determine. The problem of correcting for these

corner and edge losses, or equivalent cross sectional area, is too complicated and involves too many variables to be solved here. We may, however, arrive at some indication of the magnitude of this error from the similar but much less difficult case of a spherical box. If a spherical hot box is used whose inside radius is r_1 and whose outside radius is r_2, the heat passing through the shell will be:

$$H = \frac{cA\,(t_1 - t_2)}{d} \qquad (13)$$

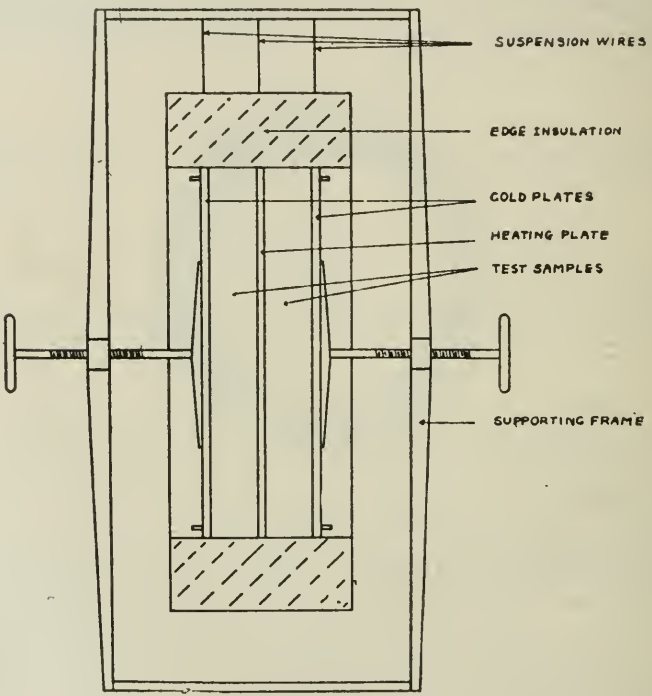

FIG. 4. DIAGRAMMATIC SKETCH OF APPARATUS FOR THE PLATE METHOD OF MEASURING THE THERMAL CONDUCTIVITY OF MATERIALS.

where
c = the conductivity
A = the area of the path
t_1 = the inside temperature
t_2 = the outside temperature
d = the thickness of the shell, or $r_2 - r_1$

If we assume the correct area to be the mean between the inner and outer surfaces, or, $\frac{4\pi r_1^2 + 4\pi r_2^2}{2}$, equation (13) becomes:

$$H = \frac{c\,(t_1 - t_2)\dfrac{4\pi r_1^2 + 4\pi r_2^2}{2}}{r_2 - r_1}$$

Transposing:

$$H = c\,(t_1 - t_2)\,4\pi\left[\frac{r_1^2 + r_2^2}{2\,(r_2 - r_1)}\right] \qquad (14)$$

Let Fig. 3 represent a cross section of the spherical box and r the radius of any spherical shell, concentric with the box, whose wall is of

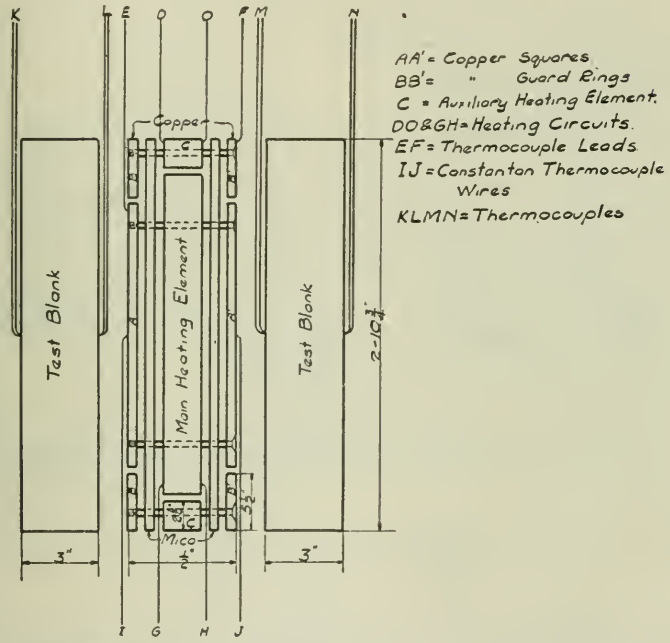

FIG. 5. DIAGRAMMATIC SKETCH OF AIR-COOLED PLATE SHOWING TEST BLANKS IN PLACE.

infinitesimal thickness dr. The temperature drop through this shell will be dt, and the temperature gradient through the shell will be $-\frac{dt}{dr}$. The heat passing through this shell will then be:

$$H = -cA\frac{dt}{dr} \qquad (15)$$

where A is the surface area of the shell or $4\pi r^2$. Substituting this value for A in equation (15):

$$H = -4c\pi r^2\frac{dt}{dr}$$

Transposing:

$$-\frac{dr}{r^2} = \frac{4c\pi\, dt}{H}$$

This being a general equation for any infinitesimal shell in the box, we may integrate between the limits of r_2 and r_1, which will give the true equation for conductivity.

$$\int_{r_1}^{r_2} -\frac{dr}{r^2} = \frac{4\ c\ \pi}{H} \int_{t_2}^{t_1} dt \qquad (16)$$

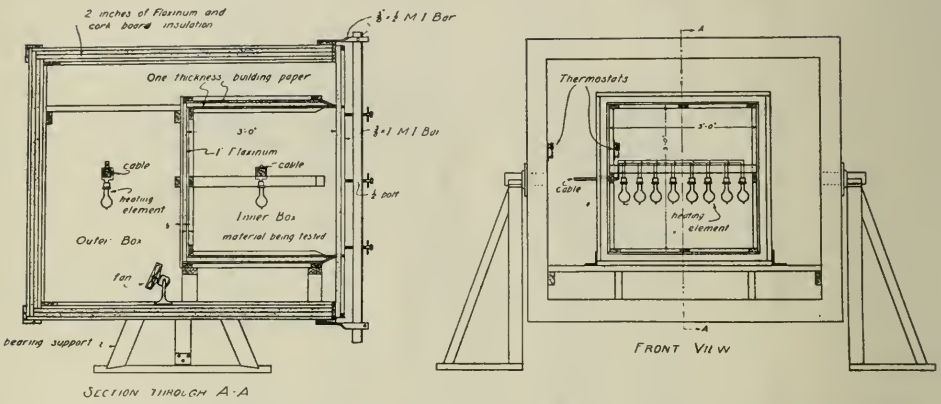

FIG. 6. SECTIONAL AND FRONT VIEWS OF THE GUARDED-AIR BOX IN USE AT THE UNIVERSITY OF MINNESOTA FOR TESTING HEAT TRANSMISSION.

$$\sum_{r_2}^{r_1} -\frac{1}{r} = \frac{4\ c\ \pi}{H} \sum_{t_2}^{t_1} t$$

$$\frac{1}{r_1} - \frac{1}{r_2} = \frac{4c\pi}{H}(t_1 - t_2)$$

$$H = c\ (t_1 - t_2)\ 4\pi \left[\frac{r_1 r_2}{r_2 - r_1}\right] \qquad (17)$$

Equations (17) and (14) are different expressions for the same thing, (17) being the result of a mathematical solution and (14) being based on an approximation. It is evident that the terms in brackets express what is expressed in equation (13) by $\frac{A}{d}$. It is also evident that the two expressions are not equal. Now if our spherical box has such dimensions that r_1 and r_2 are equal to 10 and 13 in. respectively, the spherical box will approximate the cubical box in size and we can evaluate and compare the brackets in equations (14) and (17).

Equation (14) becomes:

$$H = c\,(t_1 - t_2)\,4\pi\left[\frac{(10^2 + 13^2)}{2\,(13 - 10)}\right]$$

$$H = c\,(t_1 - t_2) \quad 4\pi \times 44.833$$

And equation (17) becomes:

$$H = c\,(t_1 - t_2)\,4\pi\left[\frac{(10 \times 13)}{13 - 10}\right]$$

$$H = c\,(t_1 - t_2) \quad 4\pi \times 43.33$$

The two factors entering into the equations differ by about 3½ per cent. Thus for the spherical box considered, the mean area will give

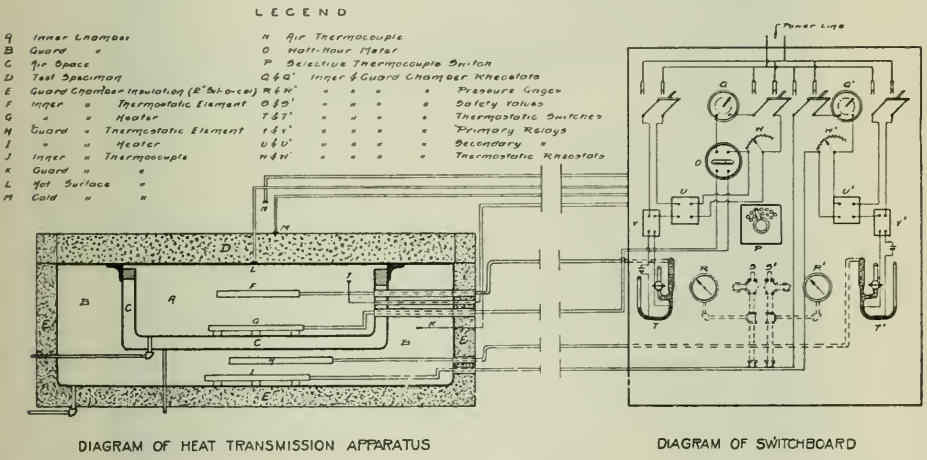
LEGEND
A Inner Chamber
B Guard "
C Air Space
D Test Specimen
E Guard Chamber Insulation (Rice)
F Inner " Thermostatic Element
G " " Heater
H Guard " Thermostatic Element
I " " Heater
J Inner " Thermocouple
K Guard " "
L Hot Surface "
M Cold " "
N Air Thermocouple
O Watt-Hour Meter
P Selective Thermocouple Switch
Q & Q' Inner & Guard Chamber Rheostats
R & R' " " " " Pressure Gauge
S & S' " " " " Safety Valves
T & T' " " " " Thermostatic Switches
U & U' " " " " Primary Relays
V & V' " " " " Secondary "
W & W' " " " " Thermostatic Rheostats
Power Line
DIAGRAM OF HEAT TRANSMISSION APPARATUS
DIAGRAM OF SWITCHBOARD

FIG. 7. DETAILS OF APPARATUS USED IN GUARDED-BOX METHOD OF DETERMINING HEAT TRANSMISSION IN USE AT THE RESEARCH LABORATORY.

an error of about 3½ per cent, which is not much greater than the homogeneity of the material. It is safe to assume that a similar error exists in the case of any other than a spherical box. We are, however, unable to infer its magnitude.

E. Hot plate method. In this method a flat hot plate, made of an electrical heating element held between two metal sheets, is the source of heat. On either side of this hot plate, specimens of the material to be tested are placed. The heat generated in the hot plate passes out through the specimens. Two water-cooled plates may be placed on the outside of the material or these surfaces may be exposed to cold air in a constant temperature room. A separate heating element called a guard heater is often used to supply the heat lost from the edges.

If the water-cooled plates are used the temperature of the hot plate and the cold plates, and the electrical input are the necessary data for determining the conductivity constants.

If the outside surfaces of the material are exposed to the air, the

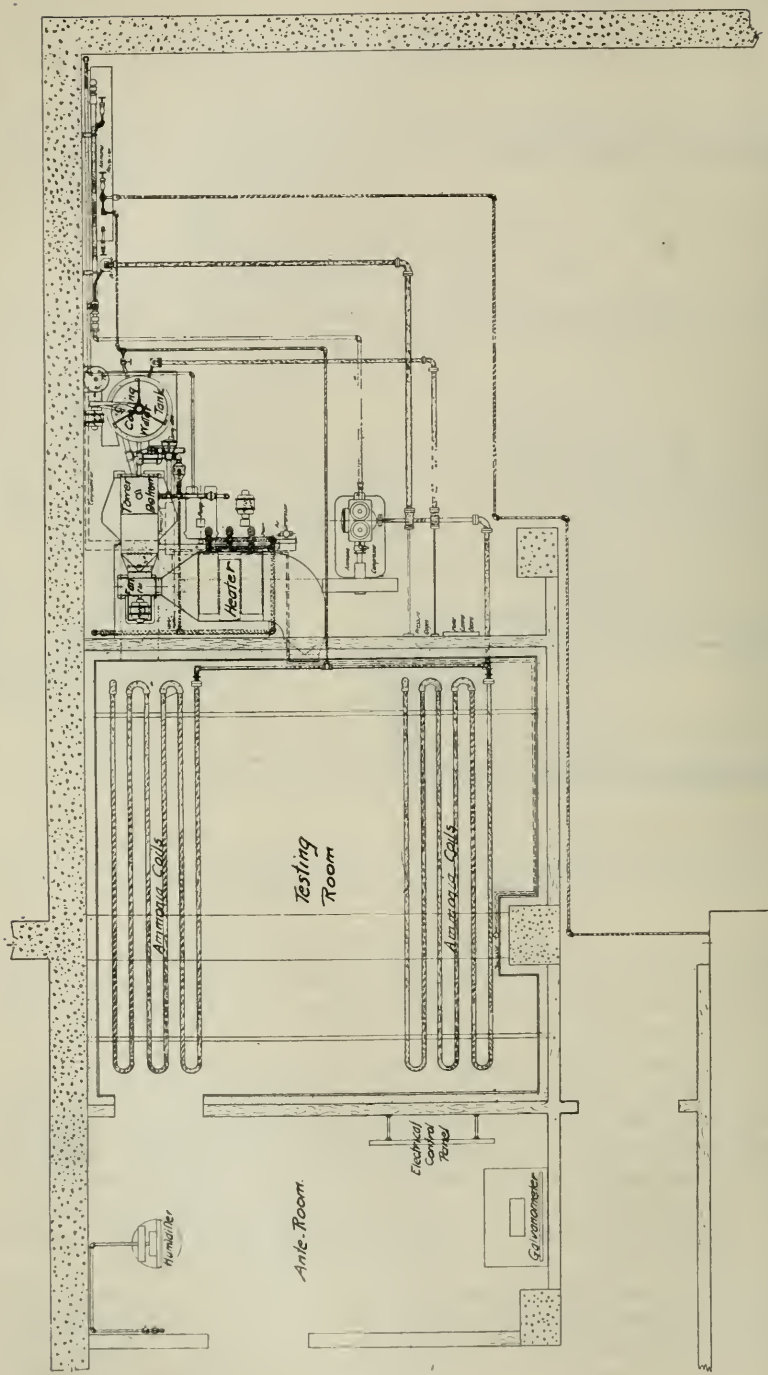

FIG. 8. LAYOUT OF APPARATUS IN THE PSYCHROMETRIC OR CONSTANT-TEMPERATURE ROOM AT THE RESEARCH LABORATORY IN PITTSBURGH.

temperature of the hot plate, the temperature of the cold surfaces of the material, the temperature of the air, and the electrical input, are the necessary data for determining the conductivity and the outside surface transmission constants.

The chief disadvantages of this method are:

1. Difficulty in sufficiently guarding against heat loss from the edge of the material. This may, however, be counteracted by making the plate of sufficient size, depending upon the thickness of the material tested.

FIG. 9. PSYCHROMETRIC OR CONSTANT-TEMPERATURE ROOM EQUIPPED WITH GUARDED-VAPOR BOX AT RESEARCH LABORATORY.

2. Difficulty in placing the test blanks of certain kinds of building material firmly in contact with the plate.

3. Two specimens of the material are necessary.

Fig. 4 is a diagrammatic sketch of water-cooled plate used by the United States Bureau of Standards. (See October, 1920, issue of the Journal, page 625.)

Fig. 5 is a diagrammatic sketch of an air-cooled plate used by the Engineering Experiment Station, Pennsylvania State College, in cooperation with the Research Laboratory of the Society.

F. Guarded box method. This method is really a hot box in which five of the sides (or one end and the cylindrical surface, if it is a cylindrical box) are guarded against the transfer of heat by maintaining constant temperatures on the two surfaces of these sides. The material to be tested either forms the sixth side (or the other end, if it is a cylindrical box) of the box or is placed on the outside of a metal forming this

side of the box. This guarded box may either contain hot air or saturated vapor of some liquid boiling at the desired temperature.

The chief disadvantages of the guarded box are:

1. Difficulty in sufficiently guarding against heat loss from the edge of the material.

2. Difficulty in sufficiently guarding all but the one side of the box against heat transfer.

3. Difficulty in constructing a leak proof box.

Fig. 6 is a drawing of a guarded air box used by the University of Minnesota. Fig. 7 is a drawing of a guarded vapor box designed and constructed by the Research Laboratory of the Society.

DISCUSSION OF RESEARCH WORK ON HEAT TRANSMISSION THROUGH BUILDING AND ALLIED MATERIALS BY VARIOUS INVESTIGATORS

Péclet in 1844 published the results of a very thorough series of investigations on heat transmission through various materials, mostly metals. Many of our modern ideas concerning the theory of heat transmission were established by him. Most of the constants used at the present time (for building material) come from his work or that of his followers, Recknagle and Rietschel.

Charles F. Hauss of Antwerp, Belgium, presented a paper before the Society in 1904 in which he gives a table of total transmission coefficients for a number of types of construction. See TRANSACTIONS, Vol. 10, p. 114.) These coefficients were not determined experimentally by Mr. Hauss. They probably come from Péclet or his followers, Recknagle and Rietschel.

L. A. Harding, while at the Engineering Experiment Station, Pennsylvania State College, built a testing plant and conducted experiments on a number of building and insulating materials. (See TRANSACTIONS, Vol. 19, p. 208, Heat Losses by Transmission Through Various Building Materials.) The hot box method was used, the heat being supplied electrically. Constants for radiation, convection, conductivity, and total transmission for a number of materials are quoted from a number of authorities.

J. A. Moyer, formerly of the Engineering Experiment Station, Pennsylvania State College, conducted some experiments on heat transmission through glass and red building brick in still and moving air. (See *A. S. R. E. Journal*, November, 1915. Also Engineering Experiment Station Bulletin 16, Pennsylvania State College.) The hot box method was used. Total transmission constants only were determined and are given for both still and moving air, of various humidities.

Dickensen and Van Dusen have conducted a large amount of research work on the conductivity of insulating materials, particularly those used by refrigerating engineers. (See The Testing of Thermal Insulators, *A. S. R. E. Journal*, September, 1916. Also Thermal Conductivity of Heat Insulators, Van Dusen, October, 1920, issue of the JOURNAL, p. 625.) A small water-cooled plate, Fig. 4, was used. Temperatures were de-

termined by means of thermocouples. Small samples only of the materials were tested and conductivity constants alone determined. A good description of their plate is given. Constants are given for some 60 insulating materials, including a few kinds of wood used in ordinary building construction.

John R. Allen, while at the University of Michigan, conducted some experiments on heat transmission through glass. (See Heat Transmission Through Building Materials, Transactions, Vol. 22, p. 507.) The hot box method was used. Five sides of the box being constructed of standard insulating material and the sixth side of the material under test. The

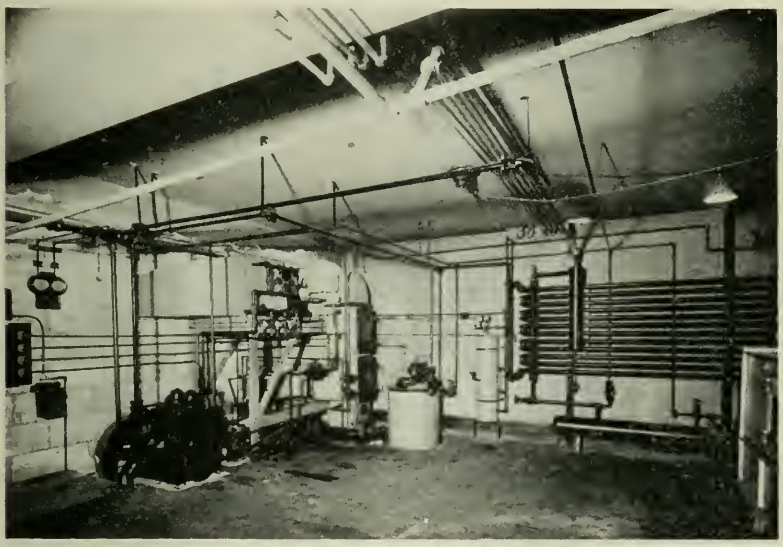

FIG. 10. TEMPERATURE, HUMIDITY AND AIR MOTION CONTROL EQUIPMENT FOR THE PSYCHROMETRIC OR CONSTANT-TEMPERATURE ROOM AT THE RESEARCH LABORATORY.

heat loss through the five sides of the box per degree temperature difference was determined. The heat loss through the test specimen was then assumed to be the difference between the loss with the test specimen in place and the loss from the other five sides as previously determined. The box itself was placed on trunnions so that the test specimen could be placed in any position. Tests were conducted with the test specimen vertical, horizontal forming the ceiling, horizontal forming the floor of the box, and at an angle of 45 deg. Tests were also conducted under conditions of wind and rain. A number of coefficients are given, some of which are repeated in Table 2.

A. C. Willard and L. C. Lichty (see Engineering Experiment Station University of Illinois, Bulletin No. 102) conducted a number of experiments on the heat transmission of various building and insulating materials using the hot box method. Large boxes were built of the materials and heated electrically. Coefficients for total transmission, conductivity, and inside and outside, surface transmission were determined for a large

number of materials. The most of these are insulating materials. Thermocouples were used in measuring surface temperatures. The effect of air motion and humidity on the transmission constants was also determined. The bulletin also contains a discussion of the simple mathematical theory of heat flow.

The Engineering Experiment Station of Pennsylvania State College, under the directions of A. J. Wood (see Bulletin No. 30), conducted experiments on the heat transmission of cork board and air spaces. Two different types of boxes were used. In one box five sides were permanently constructed of standard material and calibrated. The test specimen being placed on the sixth side. The total heat input minus that found by calibration to pass through the five sides was assumed to pass through the specimen forming the sixth side.

The other box, a cube 2 ft. on a side, was entirely built of the material tested. An exhaustive study of the two boxes was made, resulting in the conclusion that the cubical box would give consistent and satisfactory results if good judgment were used in its operation. Constants are given for cork board and air spaces. Temperatures were determined by means of electrical resistance thermometers.

The Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station of Pennsylvania State College constructed and tested a large air cooled plate for determining heat transmission constants for building materials. (See July, 1921, issue of JOURNAL, p. 529). Tests were conducted on cork board. Conductivity and surface transmission constants are given for this material. Temperature gradients over the surface of the plate were studied and temperature gradient curves away from the cork board were plotted. Temperatures were determined by copper constantan thermocouples. Work with this plate is being contributed by Prof. A. J. Wood.

F. B. Rowley, University of Minnesota, conducted some tests on insulating materials, using a guarded air box figure. (See May, 1921, issue of JOURNAL, p. 469). Temperatures were measured by means of copper constantan thermocouples. Data showing the performance of the box are given.

WORK OF THE RESEARCH LABORATORY

The policy of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in regard to the transmission of heat through building material and heat insulating material has been to follow out the program given here:

1. To survey the literature on the subject and determine what work has already been done, what methods of determining constants are most satisfactory, and what constants, if any, are acceptable as standards.

2. To develop and accept as a standard, a method for determining constants which is accurate and applicable to the materials to be tested and convenient of operation.

3. To determine and carefully check constants for a large number of building and insulating materials; these constants to be accepted as standards by the Society.

As yet no constants have been accepted by the Society. The United States Bureau of Standards has built a water-cooled plate and has determined with it, a large number of constants for insulating materials. (See October, 1920, issue of Journal, p. 625.) While this work was not conducted on a cooperative basis, the Research Laboratory was greatly interested in it.

The Research Laboratory, in cooperation with the Engineering Experiment Station of Pennsylvania State College, built and conducted tests on an air cooled plate. (See July, 1921, issue of Journal, p. 529). This plate gives considerable promise as an accurate method of determining

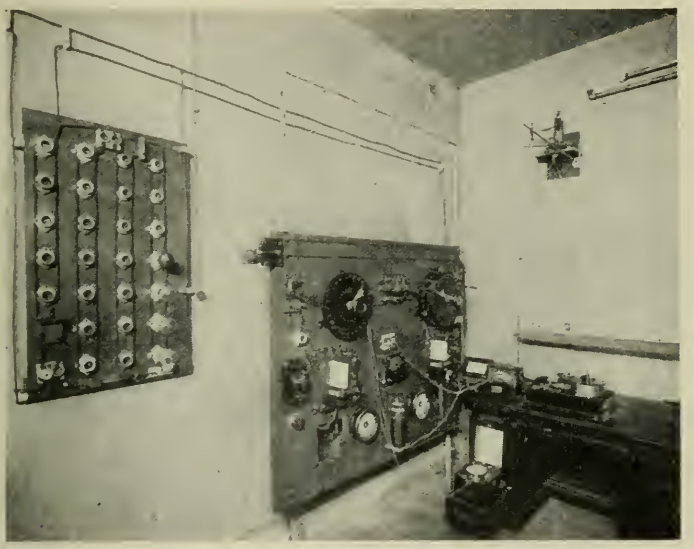

FIG. 11. TEMPERATURE MEASURING AND CONTROL EQUIPMENT FOR GUARDED-VAPOR BOX AT RESEARCH LABORATORY.

constants. It is, however, not as convenient of operation as is desired and cannot yet be accepted as the standard method of determining these constants.

A constant temperature room (Figs. 8 and 9) for heat investigations is being built at the Pittsburgh Laboratory of the United States Bureau of Mines. This room and equipment, which is now about completed, is equal to the best in the country and well fits the laboratory for heat transmission investigations. The room has no windows and is 17.5 ft. by 17.5 ft. by 10 ft. high. The entrance is by another room without windows 9 ft. by 18 ft. by 10 ft. high. The walls, floor and ceiling of the constant temperature room are lined with 4 inches of cork board and covered with hard plaster. The room may be cooled by means of a four-ton refrigerating machine or heated by means of three vento heaters over which recirculated air is blown. A humidity control equipment is also provided. The equipment (Fig. 10) and control apparatus are all located

in another room. The temperature, humidity and air motion inside the constant temperature room can therefore be controlled from without.

THE GUARDED VAPOR BOX

The Research Laboratory has designed and constructed a guarded vapor box which it is hoped will be as accurate and otherwise applicable to the determination of constants for building material as the air cooled plate, and which gives promise of being much more simple of operation. This box will be used in the new constant temperature room now being completed.

Fig. 7 is a drawing of this box. A is the inner chamber 2 ft. in diameter and 8 in. high. The inner chamber is surrounded on all sides excepting the top by the outer or guard chamber B. A and B are separated by a 1 in. air space C. The blank, D, to be tested, is placed on top of the two chambers. The sides and the bottom of the guard chamber are insulated with 2 in. of sil-o-cel powder. About three inches of aqua ammonia, whose concentration is such that it will boil at the desired temperature, is contained in each chamber. The temperature of the inner chamber is maintained constant by means of an electrical heater G, which is regulated by a thermostat F. J is a thermocouple indicating the temperature of the inner chamber. The guard chamber also contains an electrical heater I, thermostat H, and thermocouple K.

The box is constructed of ¼-in. boiler plate. Great difficulty was experienced in the construction of this box in providing leak-proof connections for the heaters, thermostats and thermocouples. In fact after receiving the box from the maker (the Reliance Boiler Works of Pittsburgh) and installing the instruments ready for test, it was found that there were several leaks between the inner and guard chambers. As such a transfer of vapor from one chamber to the other is accompanied by a large transfer of heat the box could not be used in this condition. It was necessary, therefore, to return the box to the makers. After practically reconstructing the box the instruments have again been installed with certain modifications resulting from past experience. No leak has manifested itself yet. In repairing the box the top has warped somewhat due to acetylene welding.

SOURCES OF ERROR IN THE USE OF THE GUARDED VAPOR BOX

The following possible sources of error must be considered in the use of the guarded box:

1. Unequal heating of the top surface of the box due to temperature gradient in the ammonia vapor. These surfaces are heated by wet ammonia vapor which should give a very uniform temperature. There will, however, be slight temperature gradients in the vapor, the magnitude of which must be determined experimentally since no one has been found who will predict what it will be.

2. Lateral flow of heat through the specimen D due to loss of heat from the edge. This loss is identical with the edge loss in the case of the hot plate method and depends upon the conductivity, area and thickness of the material under test. Since the area of the specimen is determined

by the size of the box, the thickness of the material to be tested must necessarily be limited. The maximum thickness of any given material to be used must be determined experimentally. Obviously the thickness must be such that the departure of the lines of heat flow from straight lines is not great.

3. Transfer of heat through the partition separating the two chambers due to a difference of temperature in the two chambers. The two chambers will be maintained at the same temperature as nearly as possible. If it were possible to keep them at exactly the same temperature there would be no transfer of heat between them. This can be done, however, only to within certain limits. The partition separating the two chambers is made up of the following parts: 1-1/3 sq. ft. of steel and 9.2 sq. ft. of two thicknesses of ¼-in. steel separated by one inch of air. The total

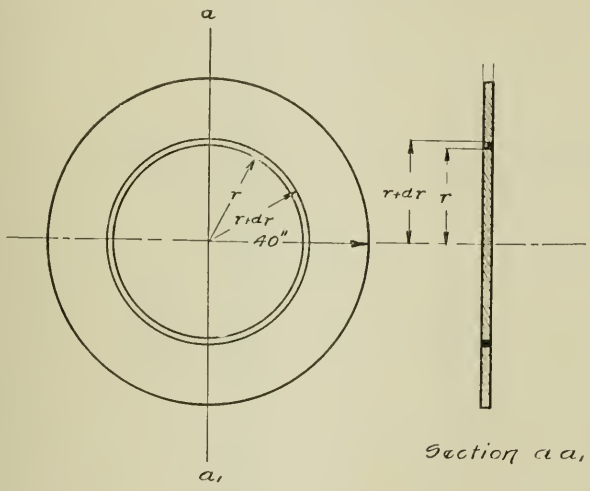

FIG. 12. TOP PLATE OF GUARDED-VAPOR BOX.

transmission of heat through the steel (wet vapor to dry vapor; the vapor in the chamber receiving the heat may be considered dry since this surface of the partition will be at a higher temperature than the vapor) will be 4 B.t.u. per sq. ft. per deg. temperature difference or 5.3 B.t.u. per degree temperature difference for the 1-1/3 sq. ft. The total transmission through the two ¼-in. thicknesses of steel plate separated by 1-in. air space is 0.25 B.t.u. per sq. ft. per deg. temperature difference or 2.3 B.t.u. per deg. temperature difference for 9.2 sq. ft. The total heat transfer through the partition between the two chambers per degree temperature difference will therefore be 7.6 B.t.u.

4. Loss of heat from the center section of the top (over the inner chamber to the outer section of the top, over the guard chamber) or vice versa due to a difference of temperature between the two chambers.

The top of the box is a lamina or, cylinder 40 in. in diameter and ¼ in. in length. We wish to determine the heat flowing across a cylindrical surface in this cylinder whose radius is r.

The lateral flow of heat from the center of the cylinder is given by

$$H = cA\frac{dt}{dr} \tag{18}$$

where c is the conductivity of the iron in B.t.u. per sq. in. per deg. temperature difference per in. thickness. A the area of the path and $\frac{dt}{dr}$ the temperature gradient.

Consider the flow of heat across any cylindrical shell whose inside radius is r and whose thickness measured radially is dr (Fig. 12).

Then $A = 2\pi rd$, where d is the length of the cylinder, or ¼ in. in our case. Equation 18 then becomes

$$H = \frac{1}{2}\pi cr\frac{dt}{dr}$$

$$\frac{dr}{r} = \frac{1}{2}\pi\frac{c}{H}dt \tag{19}$$

Integrating between the limits r_2 and r_1, and t_1 and t_2.

$$\int_{r_1}^{r_2}\frac{dr}{r} = \frac{1}{2}\pi\frac{c}{H}\int_{t_2}^{t_1}dt \tag{20}$$

$$\sum_{r_1}^{r_2}\log r = \frac{1}{2}\pi\frac{c}{H}\sum_{t_2}^{t_1}t \tag{21}$$

$$(\log r_2 - \log r_1) = \frac{1}{2}\pi\frac{c}{H}(t_1 - t_2) \tag{22}$$

$$H = \frac{1}{2}\pi c\frac{t_1 - t_2}{\log\frac{r_2}{r_1}} \tag{23}$$

Substituting the limits

$r_2 = 13$ in.
$r_1 = 12$ in.
$t_1 - t_2 = 1$ deg. fahr.
$c = 2.5$ B.t.u.

$$H = \frac{2.5 \times 3.14}{2} \times \frac{1}{.077} = 50.97 \text{ B.t.u.}$$

STANDARD THERMAL RESISTANCES

By means of the guarded box it is planned to calibrate some standard resistances to be used later in determining the conductivity of materials. or the transmission through walls already built. This idea is one which the writer discussed with the late J. R. Allen, who commended it as

offering considerable promise. Percy Nichols, at the Semi-Annual Meeting of the Society, 1921, held at Cleveland, suggested the desirability of establishing such heat resistance standards. The idea has since been used by Dr. G. R. Greenslade of the United States Bureau of Mines in determining the heat loss through a tunnel wall. The method is similar to that used in electrical measurements where the standard electric resistance is one of the two standards on which all electrical measurements are based. A thin slab of some material with thermocouples permanently located in the two faces is calibrated to give heat flow per sq. ft. of cross section with any given temperature drop between its two faces.

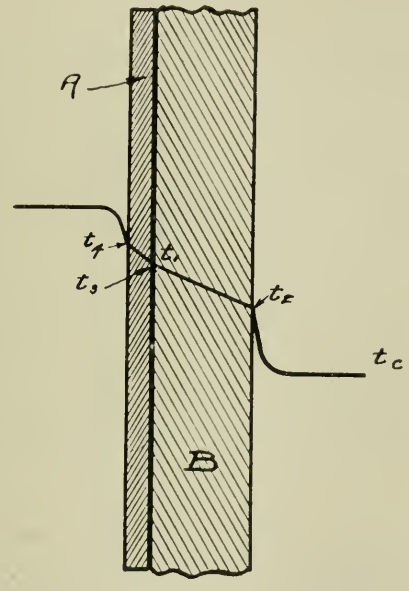

FIG. 13. USE OF STANDARD THERMAL RESISTANCE IN DETERMINING HEAT LOSSES THROUGH WALLS.

If such a thin slab A (Fig. 13) is placed against another piece of material or wall B and the temperature drop $(t_4 - t_3)$ through the calibrated slab and the temperature drop $(t_1 - t_2)$ through the wall are determined the conductivity of the section B may be calculated as follows:

The quantity of heat flowing through the calibrated slab A per sq. ft. may be taken from a curve. It represents also the heat passing through each sq. ft. of the wall B. The heat passing through each sq. ft. of the wall B is given by H in the following equation:

$$H = \frac{c}{d}(t_1 - t_2)$$

The conductivity of the material is then given by c in the following equation:

$$c = \frac{Hd}{t_1 - t_2}$$

Standard resistance for this purpose must be made of materials whose conductivity does not change with time and which are not affected by ordinary changes in relative humidity. They must be large enough in area to cover a sufficient amount of the wall to insure parallel lines of heat flow at the center where the temperatures are determined. Dr. Greenslade in determining the heat loss through a tunnel wall used a sheet of lead 1 in. in thickness and 24 in. in diameter. Where the quantity of heat passing through the material is great enough, a metal is an ideal substance to use excepting for its weight. The quantity of heat flowing through most walls is not great enough to give a sufficient temperature drop through a thin metal standard, therefore it cannot be used for our work.

Some of the materials which offer considerable promise for resistance standards to be used in connection with building materials are transite board, vulcanite, rubber, fiber, and bakelite.

TABLE 2. LOSS OF HEAT THROUGH GLASS.

Quiet condition, no wind or rain	0.641
Rain, no wind	1.248
Wind, no rain	1.05
Rain and wind	1.485

From tables by John R. Allen (see TRANSACTIONS, Vol. 22, p. 525).

DISCUSSION

THORNTON LEWIS. Has the Research Laboratory approved Table 2, Loss of Heat Through Glass, in Mr. Houghten's paper, or is it simply republished from Prof. Allen's previous measurements without comment?

THE AUTHOR. They are only republished. We make no comment on their accuracy or value.

F. PAUL ANDERSON. I hope the members of this society will begin to use this cooling chart with as much freedom and with as much surety as they have been using the psychrometric chart. We will be very glad to have this become a tool for the heating and ventilating engineer to use and make a part of his daily activities. If there is one piece of research of which we feel quite proud, it is the work that has been done in the development of this chart. It is going a little bit further than the scope of the Research Laboratory. It is really a piece of physical research, and I am very much gratified that this society has been responsible for an investigation of this kind. In my judgment if the laboratory had done nothing else since its creation than produce this chart, all the money that has been expended on the laboratory has been worth while.

No. 620

TEMPERATURE, HUMIDITY AND AIR MOTION EFFECTS IN VENTILATION

By O. W. Armspach and Margaret Ingels, Pittsburgh, Pa.

Members

IN this paper the fundamental laws underlying the loss of heat from the human body and the resulting feelings of comfort or discomfort in air under various conditions of temperature, humidity and rate of air movement will be considered. It is commonly accepted by engineers and hygienists, and there is much experimental evidence in proof of the same, that the comfort or discomfort of the body, ability to work at maximum efficiency, and resistance against disease, depend, in large measure, on the temperature, humidity and rate of air movement. The increasing importance given to these three factors in ventilation is evidenced by the numerous investigations being made at the present time, particularly in connection with the ventilation of working places in coal and metal mines. Considerable progress has been made to the extent that certain combinations of temperatures, humidities and air velocities have been found comfortable while other combinations have caused a rise in body temperature and a decrease of 25 per cent to 75 per cent in the efficiency of workers, but no complete and consistent data for all conditions encountered are available.

Although the internal temperature of the human body is substantially constant it is not absolutely so. It will vary slightly with the external conditions and there is a certain relationship between the body temperature, skin temperature and the temperature, humidity and velocity of air surrounding the body. A change in the physical condition of the air will cause a change in skin temperature and also a change in the heat loss from the body. The writers are of the belief that skin temperature ultimately determines the individual's comfort or discomfort since it is this temperature which reacts on the heat controlling mechanism of the body. However, the variations in the temperature of the skin are so small under ordinary conditions that the rate of heat loss from the body, and therefore its degree of comfort can best be determined by observing the changes in the physical conditions of the air itself.

Since the comfort of the body depends upon its rate of heat loss it may be well first to briefly review the laws governing the means of heat elimination. Body heat is constantly generated by chemical and physical processes within. To maintain a constant temperature, heat must also be eliminated in proportion to the amount produced. This elimination con-

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1922.

tinues by radiation to surrounding walls and objects, by convection due to temperature difference, and by evaporation of moisture from the body surface. The factors which determine the rate of heat elimination by these three means are as follows:

Radiation—(a) Temperature of surrounding walls and objects; (b) Amount of window area; (c) Proximity and temperature of radiators, cooling coils and other high or low temperature sources of heat; (d) Clothing; (e) Temperature and humidity of air.

Convection—(a) Dry bulb temperature; (b) Rate of air motion; (c) Condition and amount of clothing; (d) Relative humidity.

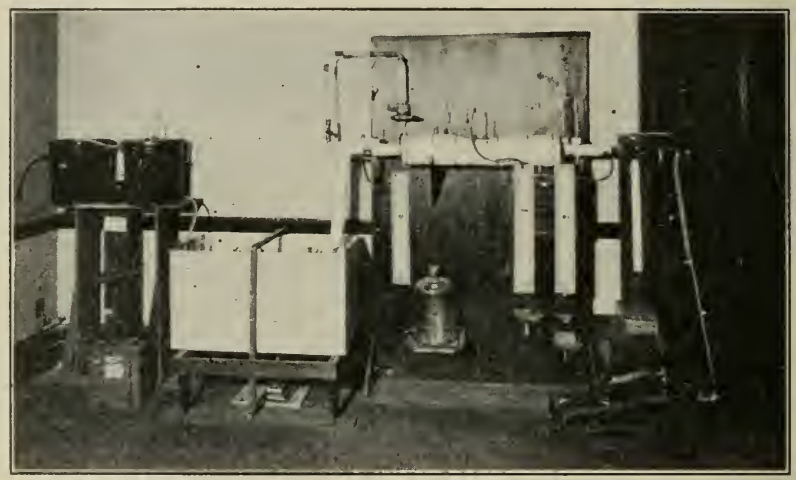

FIG. 1. APPARATUS DEVELOPED FOR ACCURATELY VARYING THE TEMPERATURE, HUMIDITY AND VELOCITY OF AIR.

Evaporation—(a) Dry bulb temperature and relative humidity; (b) Wet bulb temperature; (c) Rate of air movement; (d) Clothing.

All the foregoing factors have an effect upon the individual's health and comfort and still others, although negligible, might be mentioned. This explains in part why it is exceedingly difficult to obtain consistent opinions of comfort from different persons exposed to the same air conditions. To the extent that these factors affect the skin temperature of different persons, to just that extent will opinions differ. However, experiments which will practically eliminate this equation of personal opinion are being made to determine the true relative comfort of different atmospheres, but a discussion of these experiments will be left for another paper.

The ventilating engineer is not directly concerned with the relative proportions of heat lost by each of the three means since it is the total heat loss which determines comfort. The factors which determine the total heat loss by radiation, convection and evaporation overlap considerably. For instance, when the temperature and humidity of the air changes,

the rate of evaporation also changes and therefore the surface temperature of the body is affected. This in turn will affect the loss by radiation and convection. Fortunately, however, we have found that there is a balance between all of these effects and the equations for total heat loss are comparatively simple.

After studying the factors which affect the means of heat elimination from the human body it is apparent that the controlling factors determining comfort are the temperature of the air, relative humidity and rate of air movement. The old chemical theory, namely, that inadequate ventilation was due to some mysterious poison given off in the expired

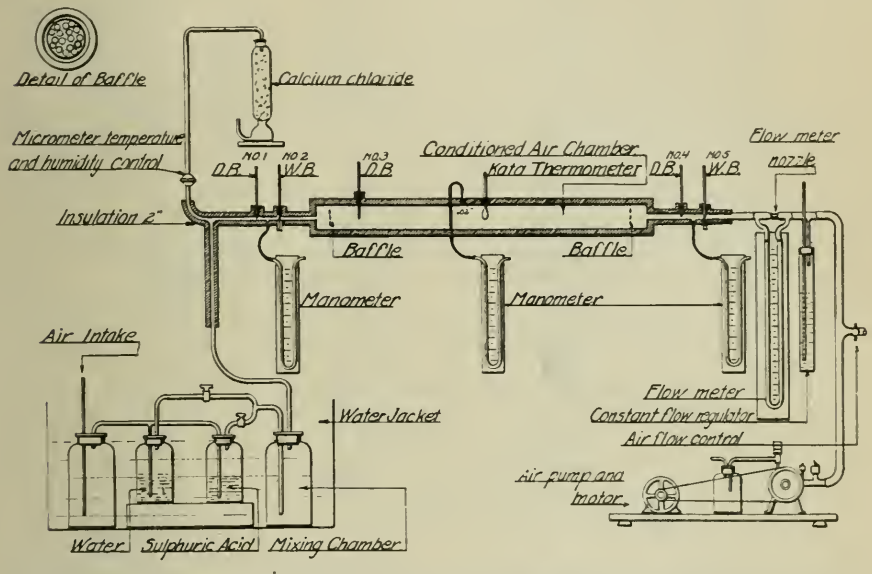

FIG. 2. APPARATUS FOR CONTROLLING TEMPERATURE, HUMIDITY AND VELOCITY OF AIR.

air with an accompanying diminution in oxygen and an increase in carbon dioxide, has apparently been replaced by the almost universal acceptance of the greater importance of these physical properties of air. By this statement we do not mean to minimize the importance of other factors in ventilation but rather to emphasize the importance of those considered in this paper.

Dr. Leonard Hill has forcibly demonstrated the correctness of the present theory of ventilation by a well known experiment. Several subjects were enclosed in an air-tight chamber of small capacity until the CO_2 content increased to nearly 4 per cent and the oxygen content decreased to nearly 19 per cent. The subjects were surprised to find that they could not light their cigarettes due to the low oxygen content of the air. They suffered great discomfort. The pulse rate increased, faces were flushed, and they perspired freely. The same air was then set in motion by means of fans located in the ceiling, and the discomfort of

the subjects immediately disappeared. The pulse rate became normal. Those on the outside of the chamber breathing the stale air through tubes leading from the inside experienced no discomfort, and those on the inside breathing pure air from the outside experienced no relief until the fans were started and the air was set in motion. This experiment, perhaps more than any other, has demonstrated the necessity of determining the relationship between temperature, humidity and air velocity based on the feelings of comfort or discomfort. There can be little doubt that

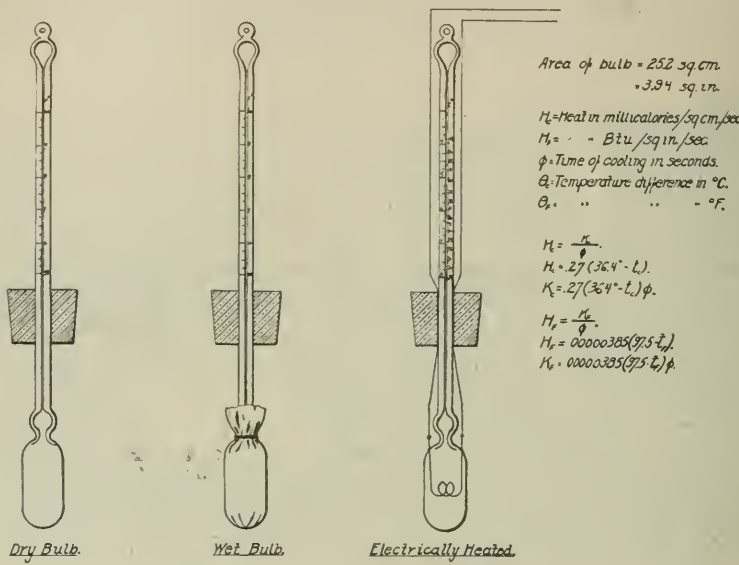

FIG. 3. THE KATA THERMOMETERS USED IN THE TESTS.

such determinations of correct air conditions will have more to do than any other factor with the maintenance of health, increase in efficiency, and the reduction of fatigue of our industrial workers.

TEST APPARATUS

To carry out the experiments described in this paper, an apparatus (Fig. 1) has been developed which enabled us to accurately vary and control the temperature, humidity and velocity of the air (Fig. 2). Air entering the equipment through a ¾ in. glass tube passes into a bottle submerged in water. From this bottle the air passes either through sulphuric acid or water. When all the air was passed through the sulphuric acid a very low humidity was obtained due to the moisture absorbing properties of the acid, but if this line was closed so that the air passed through the bottle containing water practically 100 per cent relative humidity was obtained. Two stop cocks, controlling the relative proportion passing through either solution, were used to obtain the desired humidity. The water covering this part of the apparatus could be cooled or heated as required. The micrometer temperature and humidity control

was used to allow a small portion of air dried with calcium chloride or room air at a higher temperature to pass into the system. This adjustment would vary the wet and dry bulb temperatures to a very small fraction of a degree.

The conditioned air chamber consists of a large brass tube having a uniform internal diameter of 3.871 inches and a length of 30 in. Baffles made by punching a large number of 1/8 in. holes in thin brass plates were placed near either end of tube to give a uniform velocity over the cross section. This chamber was covered with 2 inches of hair-felt insulation so as to hold the walls of the tube at the same temperature as the air within. Static tubes connected to suitable water columns gave

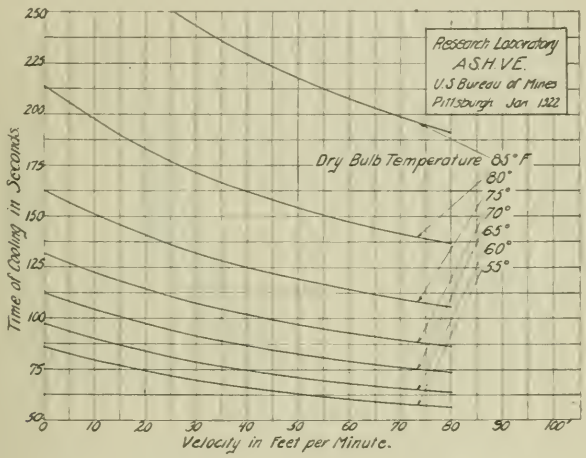

FIG. 4. EFFECT OF VELOCITY ON COOLING TIME OF DRY KATA.

the pressure of the air at three points in the apparatus. Thermometers No. 1 and No. 2 indicated the wet and dry bulb temperatures of the air entering the chamber; thermometer No. 3 the dry bulb temperature in the chamber, and thermometers No. 4 and No. 5 the wet and dry bulb temperatures of the air leaving the chamber.

Previous to taking observations the apparatus was operated until all conditions reached a constant point. A flow meter connected to a water column indicated the quantity of air passing through the system. This flow meter contains a nozzle carefully calibrated by means of a dry meter. The cubic feet of air handled was indicated on a scale to 1/1000 cu. ft. per min. An overflow placed between the pump and flow meter reduced variation in velocity to a minimum. The velocity in the chamber was obtained by dividing the cubic feet of air handled per minute by the internal tube diameter.

THE KATA THERMOMETER

The Kata thermometer was used to obtain all of the experimental data recorded in this paper (Fig. 3). This instrument, an invention of

Leonard Hill, was designed to measure its own rate of cooling in various atmospheres when its temperature was approximately the same as the skin temperature of clothed human body. The instrument is essentially an alcohol thermometer with a bulb approximately 5/8 in. in diameter and 1/2 in. long, with a stem 8 in. long reading from 100 deg. to 95 deg. fahr., graduated to tenths of a degree. To take observations the bulb is heated in water until the alcohol expands and rises into the top reservoir. It is then thoroughly dried and all traces of moisture are removed with a soft cloth. The time required for the liquid to fall from 100 deg. to 95 deg. is recorded with a stop watch and this time is a measure of the rate

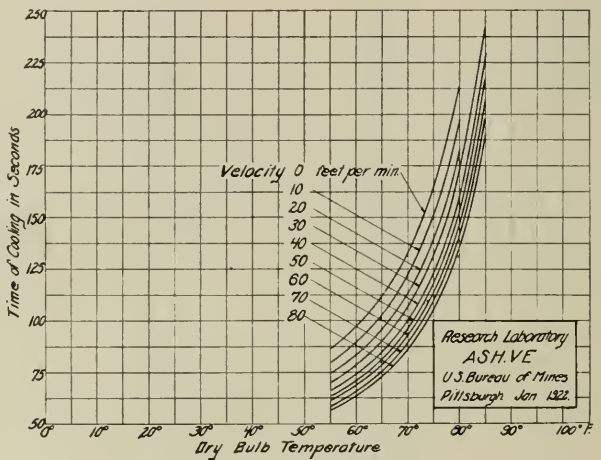

FIG. 5. COOLING OF DRY KATA IN MOVING AIR.

of cooling. It is a measure of the cooling power of the air on the dry Kata. Heat is lost from the dry Kata by radiation to surrounding walls and objects, and by convection. Dry readings, therefore, will vary with the dry bulb temperature and velocity of the air passing over the bulb.

To take wet Kata readings a cotton lisle wick is fitted snugly around the bulb. After being heated in water until the liquid rises into the upper reservoir the surplus water is removed from the bulb by being lightly brushed with a soft cloth. The time of cooling from 100 degrees to 95 degrees is again recorded as with the dry Kata and this time is a measure of the cooling power of the air on the wet surface. Heat is lost from the wet thermometer by radiation, convection and evaporation. Its readings, therefore, respond to changes in the temperature, humidity and velocity of the air passing over the bulb.

It should be observed at the outset that the factors which affect the heat loss from the wet and dry Kata are identical to the factors which affect the heat loss from the body. To the degree that these factors affect the heat loss from the body, so they also affect the comfort of the body, and it was the primary purpose of this investigation to determine the relative influence of each of these factors.

METHOD OF MAKING TESTS

With the apparatus described (Fig. 2) we were able to determine the rate of cooling of both the wet and dry Kata for many combinations of temperatures and humidities with velocities from zero to 80 ft. per min. Each series of tests was made with a constant dry and wet bulb temperature and with velocity increments of 10 ft. per min. Other groups were then made with the same dry bulb but different wet bulb temperatures and for all velocities. In like manner this procedure was repeated with other dry bulb temperatures. By this method several hundred read-

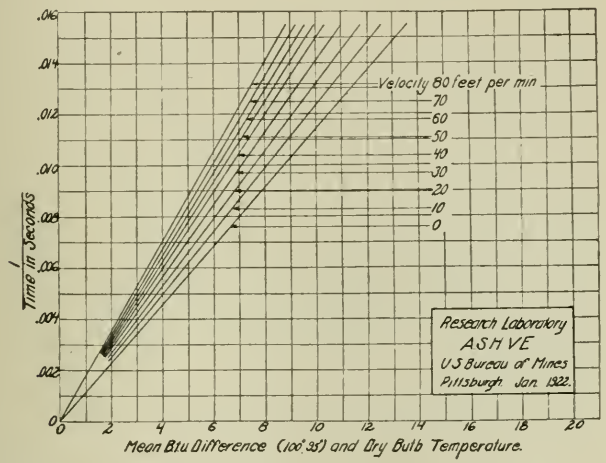

FIG. 6. THE RATE OF COOLING OF DRY KATA—PROPORTIONAL TO MEAN SENSIBLE HEAT DIFFERENCE.

ings were taken which gave the time of cooling from 100 deg. to 95 deg. for the wet and dry Kata under all possible combinations of the three influencing factors. Dry Kata readings were taken with a dry wick over the bulb for reasons which will be given later. All conditions were reduced to a standard barometric pressure of 29.92 in. of mercury.

After studying the data obtained it was found that for constant velocities the time of cooling of the dry Kata varied with the dry bulb temperature of the air; the cooling of the wet Kata varied with the wet bulb temperature of the air regardless of the dry bulb temperature or relative humidity. Hence for equal velocities:

1. The heat loss from the dry Kata is a function of the difference in sensible heat content of the air at the average temperature of the Kata (97.5 deg. fahr.) and the sensible heat content of the cooling air.

2. The heat loss from the wet Kata is a function of the difference in total heat content of air saturated at the average temperature of the Kata (97.5 deg. fahr.) and the total heat content of the cooling air.

With these laws established the experiments were repeated and all curves were checked by running the apparatus under constant wet bulb

temperatures with varying velocities. The complete results are shown in Figs. 4 to 9 inclusive.

The Research Laboratory is indebted to W. H. Carrier, who rendered available the apparatus with which we were able to determine cooling powers in air velocities as high as 2,000 ft. per min. With this apparatus, where the temperatures and humidity control were extremely accurate, we found that the same laws established for low air motions also applied at these high velocities. That is, for equal velocities the

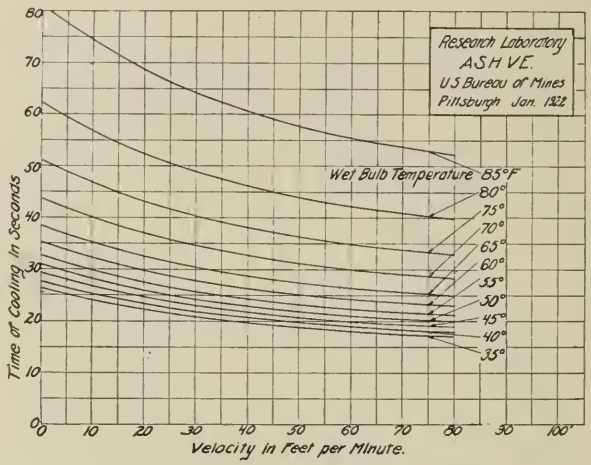

FIG. 7. EFFECT OF VELOCITY ON COOLING TIME OF WET KATA.

heat loss from the wet Kata depends only on the wet bulb temperature of the air.

HEAT CAPACITY AND THE KATA FACTOR

In order to convert the time readings obtained with the Kata to a heat basis it is necessary to determine the actual quantity of heat given up by the thermometer in dropping from 100 deg. to 95 deg. The apparatus shown in Fig. 10 was used to determine this heat capacity. A thermometer with a heating element in its bulb was inserted into the still air chamber through a hole provided in the cover. Water circulated through the apparatus held the temperature of the air at a constant point. A storage battery supplied the necessary electrical energy which was measured with an ammeter and a voltmeter. The heat loss calculated from the watts supplied was then determined for various air temperatures and this loss was plotted against the temperature difference as shown in Fig. 2. This curve shows that the heat loss from the dry Kata in still air is proportional to the temperature difference and the relation is expressed by the equation

$$H = .0000152\ (97.5 - t) \qquad (1)$$

Correctly the heat loss cannot be expressed by a single proportion since the radiation loss varies with the fourth power of the absolute tempera-

tures. However, within range of temperatures applicable to the Kata the error in assuming this simple relation is negligible.

The rate of cooling of different thermometers varies slightly and it is essential that a calibration factor be obtained so that readings taken with different instruments will give equal results under similar air conditions. This factor is called the Kata factor. Each instrument as received from the maker has this factor engraved on its stem. It is obtained by recording the item of cooling of the thermometer in still air with various temperatures. The walls of the air chamber in which the factor is determined must be at the same temperature as the air surrounding the Kata and a water jacketed or well insulated chamber is necessary to obtain an

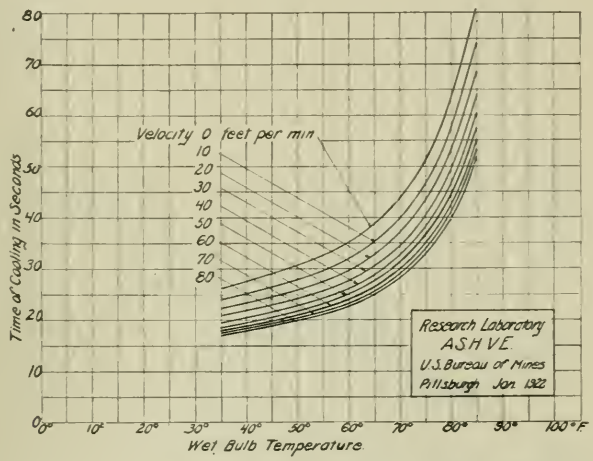

FIG. 8. COOLING OF WET KATA IN MOVING AIR.

accurate and constant result for all air temperatures. The Kata factor is calculated by multiplying the heat loss by the time of cooling in seconds.

$$K = H \times \phi \tag{2}$$

Substituting the value of H in equation (2) and dividing by 3.94 the area of the bulb in sq. in. we will have

$$K = .00000385\ (97.5 - t)\ \phi \tag{3}$$

where K = the Kata factor or the heat liberated in cooling from 100 deg. to 95 deg. per sq. in. This is a constant quantity under all conditions.

H = B.t.u. loss per sq. in. per sec. = cooling power of air.

t = dry bulb temperature of air in deg. fahr.

ϕ = time of cooling from 100 deg. to 95 deg. in seconds.

The cooling power under any condition is readily determined by dividing the Kata factor by the time.

$$H = \frac{K}{\phi} \tag{4}$$

In our experiments the value of K was equal to .01409 B.t.u. per sq. in. or 2.027 B.t.u. per sq. ft.

RESULTS OBTAINED

The wick placed around the bulb offers considerable resistance to the flow of heat from the Kata. The bare wick, of course, offers the least resistance to heat flow while wicks of different thickness will accordingly increase the resistance. Fig. 12 shows the effect of three different wicks upon the time of cooling as compared with the bare thermometer. Although the lisle wick offers the greatest resistance, it is this one which

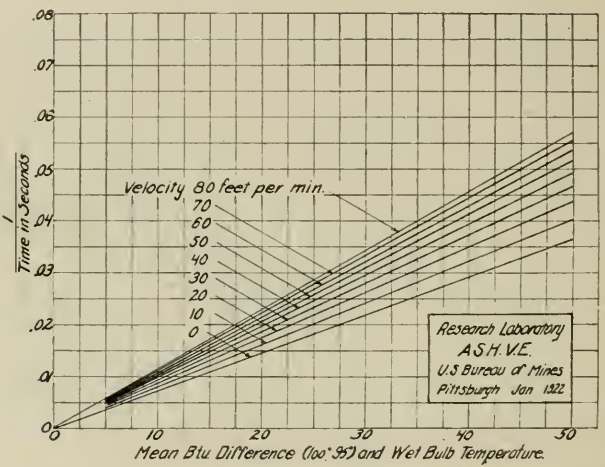

FIG. 9. THE RATE OF COOLING OF WET KATA—PROPORTIONAL TO MEAN TOTAL HEAT DIFFERENCE.

has been the one used in our experiments for the reason that its effect more nearly corresponds to the effect of clothing upon the human body. The effect of velocity on the bare thermometer is certainly much greater than the effect of velocity on the clothed human body and this is particularly true at high velocities. The effect of the wick is much more pronounced on the wet Kata, a very thin silk wick giving nearly 50 per cent greater cooling power than a cotton lisle covering. It has been stated that since heat is lost from the dry Kata by radiation and convection and from the wet Kata by radiation, convection and evaporation, then the difference between the dry and wet cooling powers would give the heat lost by evaporation. This cannot be true inasmuch as the surface conditions in the two cases are entirely different. The rate of convection and radiation is much higher with the dry wick than with the wet wick because the surface temperature is higher in the case of the dry wick. The wet surface tends to approach the wet bulb temperature of the air and under certain conditions it is possible that no heat is lost from the wet Kata by radiation or convection. To separate the cooling due to evaporation from that due to radiation and convection it would be neces-

sary to determine surface temperatures under all conditions. This could be easily done by placing thermocouples around the wet and dry Kata but this is not within the scope of this paper.

The heat loss may also be expressed as a function of the mean B.t.u. difference in sensible or total heat Figs. 6 and 9. To illustrate, assume air with a dry bulb temperature of 75 deg. and a wet bulb temperature of 58 deg.

	Dry Bulb			Wet Bulb		
	100	95	75	100	95	58
Sensible Heat (per lb. above 0° F.)	24.15	22.94	18.11			
Total Heat (per lb. above 0° F.)				68.8	60.9	24.75

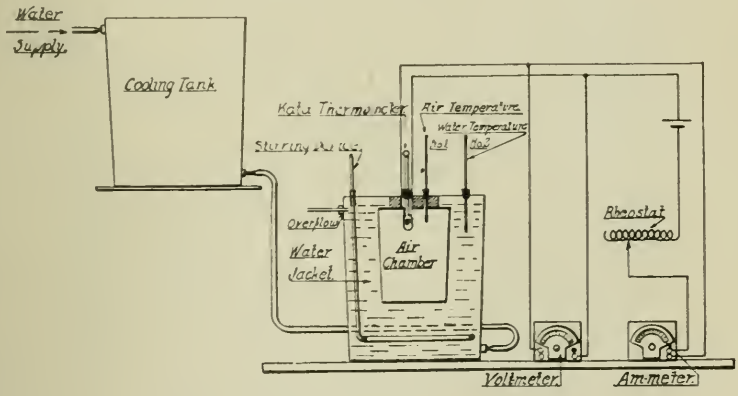

FIG. 10. STILL AIR TESTING APPARATUS.

The mean B.t.u. difference in sensible or total heat may then be calculated by the following formula:

$$(B.t.u.)_d = \frac{(B_{100} - B) - (B_{95} - B)}{log_e \dfrac{(B_{100} - B)}{(B_{95} - B)}} \qquad (5)$$

where B_{100} = B.t.u. content of air at 100 deg.
B_{95} = B.t.u. content of air at 95 deg.
B = B.t.u. content of cooling air
$(B.t.u.)_d$ = mean B.t.u. difference

If the values given in the table are substituted in equation (5), then the B.t.u. difference in total heat is 40.23 and the mean B.t.u. difference in sensible heat is 5.41. The loss per mean B.t.u. difference in sensible or

total heat content of air per sq. in. per sec. is constant for a constant velocity and may be designated by C_v. Then

$$K = C_v \ (B.t.u.)_d \phi \tag{6}$$

and

$$C_v = \frac{K}{(B.t.u.)_d \phi} \tag{7}$$

where K = B.t.u. loss per sq. in.
C_v = B.t.u. loss per sq. in. per sec. per mean B.t.u. difference.
ϕ = time of cooling in seconds.

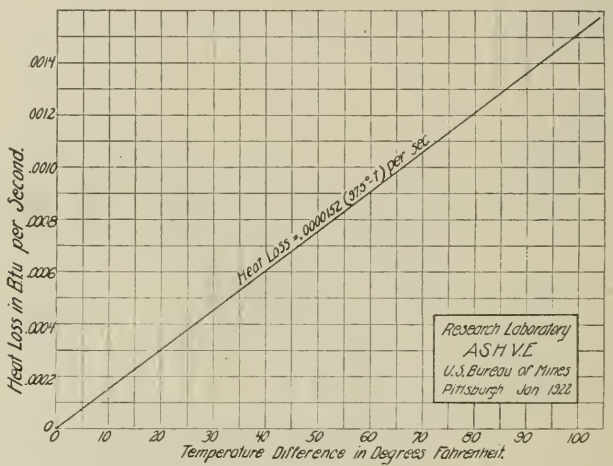

FIG. 11. MEASUREMENT OF HEAT LOSS OF KATA BY ELECTRICAL INPUT.

Fig. 13 shows the values of C_v plotted against velocity for the bare Kata, the Kata with wick and also for a pipe coil heater. Note that the curve for the bare Kata is of the same general form as the curve for the pipe coil heater. Size apparently has little influence on the convected heat loss. The difference in the two curves is probably entirely a surface effect. The heater curve necessarily passes through zero since under test conditions a correction was made for the radiation loss by eliminating end sections in the calculations. With the Kata, however, there is considerable loss by radiation and convection even at low velocities, and hence the curve does not pass through zero. There is a rapid falling off of the curve for the dry Kata with a wick, however. At low velocities its heat loss is more rapid than the loss from the heater but the curve flattens out rapidly with increased velocity evidently due to the high resistance offered by the wick. It is for this reason that the Kata with a wick is more suitable than the bare Kata as an index of the rate of heat loss from the clothed body.

Fig. 14 gives the rate of heat loss from the wet Kata in B.t.u.'s per sq. ft. per hr. per mean B.t.u. difference in total heat for various

velocities. Note that this curve, like the curve of the dry Kata with wick in Fig. 13, also falls off rapidly after a velocity of 200 ft. per min. is reached due to the high resistance of the wick and water film. This is a significant fact inasmuch as it has a direct bearing on experiments which are now being made to determine the effects of various air conditions on the human body. In these experiments, unlike experiments with the Kata, we have neither the perfectly wet nor the perfectly dry surface. The curve for the human body will probably fall somewhere between the curve for the wet and dry Kata.

FORMULAE FOR COOLING POWERS OF AIR

Designate the intercepts with the zero axis of the curves in Fig. 13 and Fig. 14 by the letter a, and let C_v be the B.t.u. loss per sq. ft. per

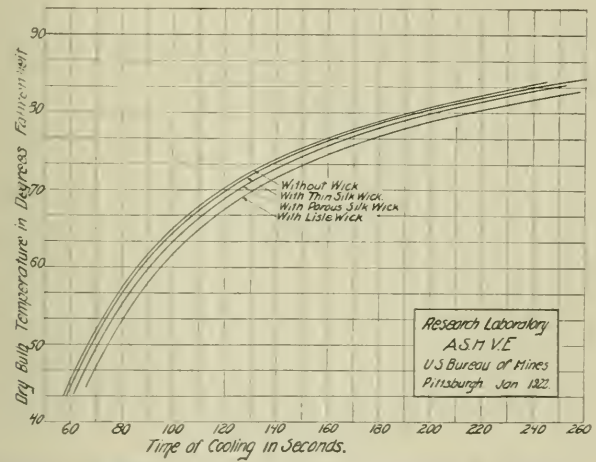

FIG. 12. EFFECT OF WICK ON DRY KATA IN STILL AIR.

hour per mean B.t.u. difference in sensible or total heat. Then, plot $\frac{1}{C_v - a}$ against the reciprocal of the velocity and we will have the straight line curves in Fig. 15 and Fig. 16. The equations of the curves may now be readily determined by finding the slopes of the lines. Let b be the intercepts at $\frac{1}{V} = 0$.

Then

$$\frac{\frac{1}{C_v - a} - b}{\frac{1}{V}} = S \qquad (8)$$

Simplifying

$$C_v = \frac{V}{S + V\,b} + a \tag{9}$$

where S = slope of line.

Substituting the numerical values taken from the curves we have, for the dry Kata

$$C_v = \frac{V}{15.9 + 0.0386\,V} + 8.25 \tag{10}$$

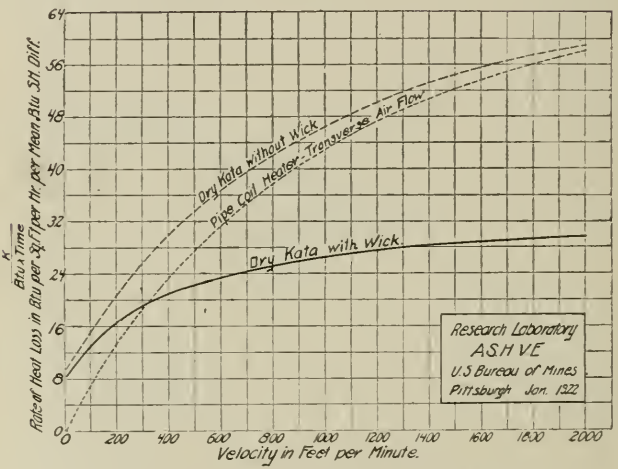

FIG. 13. RELATION OF VELOCITY TO RATE OF HEAT LOSS.

and, for the wet Kata

$$C_v = \frac{V}{19.86 + 0.0172\,V} + 5.34 \tag{11}$$

since

$$C_v = \frac{144\,K}{\dfrac{(B.t.u.)_d\ \phi}{3600}} \tag{12}$$

and

$$H = \frac{K}{\phi}$$

then

$$C_v = \frac{518{,}400\,H}{(B.t.u.)_d} \tag{13}$$

therefore

$$H_{DB} = \left[\frac{V}{15.9 + 0.0386\,V} + 8.25 \right] (B.t.u.)_s \tag{14}$$

and

$$H_{WB} = \left[\frac{V}{19.86 + 0.0772\,V} + 5.34 \right] (B.t.u.)_t \tag{15}$$

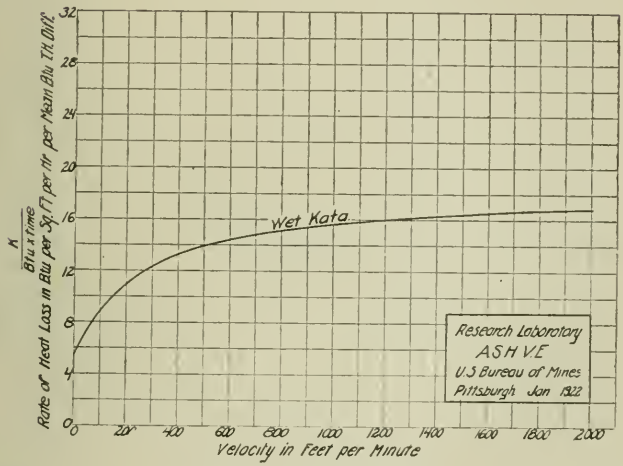

FIG. 14. RELATION OF VELOCITY TO RATE OF HEAT LOSS

where H_{DB} = dry cooling power in B.t.u. per sq. ft. per hr.
H_{WB} = wet cooling power in B.t.u. per sq. ft. per hr.
$(B.t.u.)_s$ = mean sensible heat difference.
$(B.t.u.)_t$ = mean total heat difference.
V = velocity in ft. per min.
H = cooling power in B.t.u. per sq. in. per sec.
ϕ = time of cooling in sec.
C_v = B.t.u. loss per sq. ft. per hr. per mean B.t.u. difference.
K = Kata factor in B.t.u. per sq. in. per sec.

THE COOLING POWER CHART

The chart (Fig. 17) constructed from the data obtained in this paper, shows the cooling power of air in B.t.u. per sq. ft. per hour under various conditions of temperature, relative humidity and velocity. In reading the chart the reader should keep in mind that the cooling power on the wet Kata depends on the wet bulb temperature of the air, and the cooling

power on the dry Kata varies with the dry bulb temperature of the air. Examples will illustrate the use of the curves.

Example (1). Assume air moving at a rate of 50 ft. per min. with a dry bulb temperature of 70 deg. and a wet bulb temperature of 58 deg. fahr. Required, the cooling power of this air on the wet Kata. Locate the intersection of the 70 deg. dry bulb line with the 58 deg. wet bulb line. From this point pass diagonally along the 58 deg. wet bulb line to the saturation or 100 per cent relative humidity line. Then pass vertically upward to the 50 ft. per minute wet velocity line and then horizontally to the scale on the left, and read 296 B.t.u. per sq. ft. per hr.

Example (2). Assume air at 65 deg. dry bulb and 50 per cent relative humidity moving at a rate of 50 ft. per min. Required, the cooling power

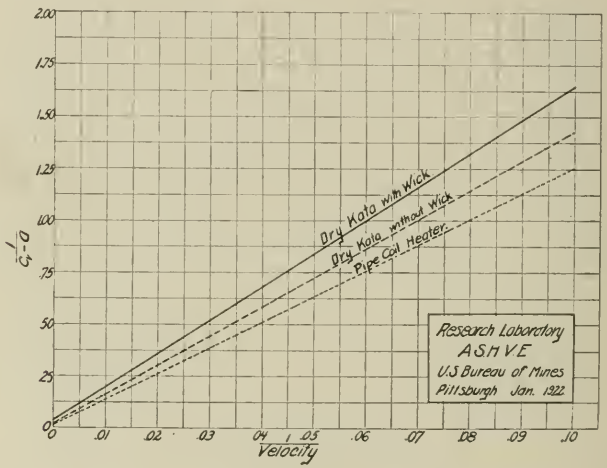

FIG. 15. DETERMINATION OF RELATION OF VELOCITY TO RATE OF HEAT LOSS.

of this air on the dry Kata. Locate the 65 deg. dry bulb line and pass vertically upward to the 50 ft. per min. dry velocity line. Then pass horizontally to the scale on the left and read 87 B.t.u. per sq. ft. per hr.

Example (3). Readings taken with the psychrometer in a certain industry indicated that the dry bulb temperature of the air was 80 deg. and the wet bulb temperature was 70 deg. The air motion was barely noticeable and owing to these oppressive air conditions the workmen were very inactive. Assume that the wet Kata cooling power of air necessary for comfort is 263 B.t.u. per sq. ft. per hr. How may the foregoing condition be remedied so that the workmen will be more comfortable? A wet bulb temperature of 70 deg. and a dry bulb temperature of 80 deg. with no air motion will give a wet Kata cooling power of 166 B.t.u. Hence, to give the correct conditions it is necessary to increase the cooling power from 166 B.t.u. to 263 B.t.u. This can be done either by lowering the temperatures or by increasing the air motion. The desirable velocity is found by locating the point of intersection of the 0 wet velocity line with the 166 B.t.u. line. Then pass vertically down-

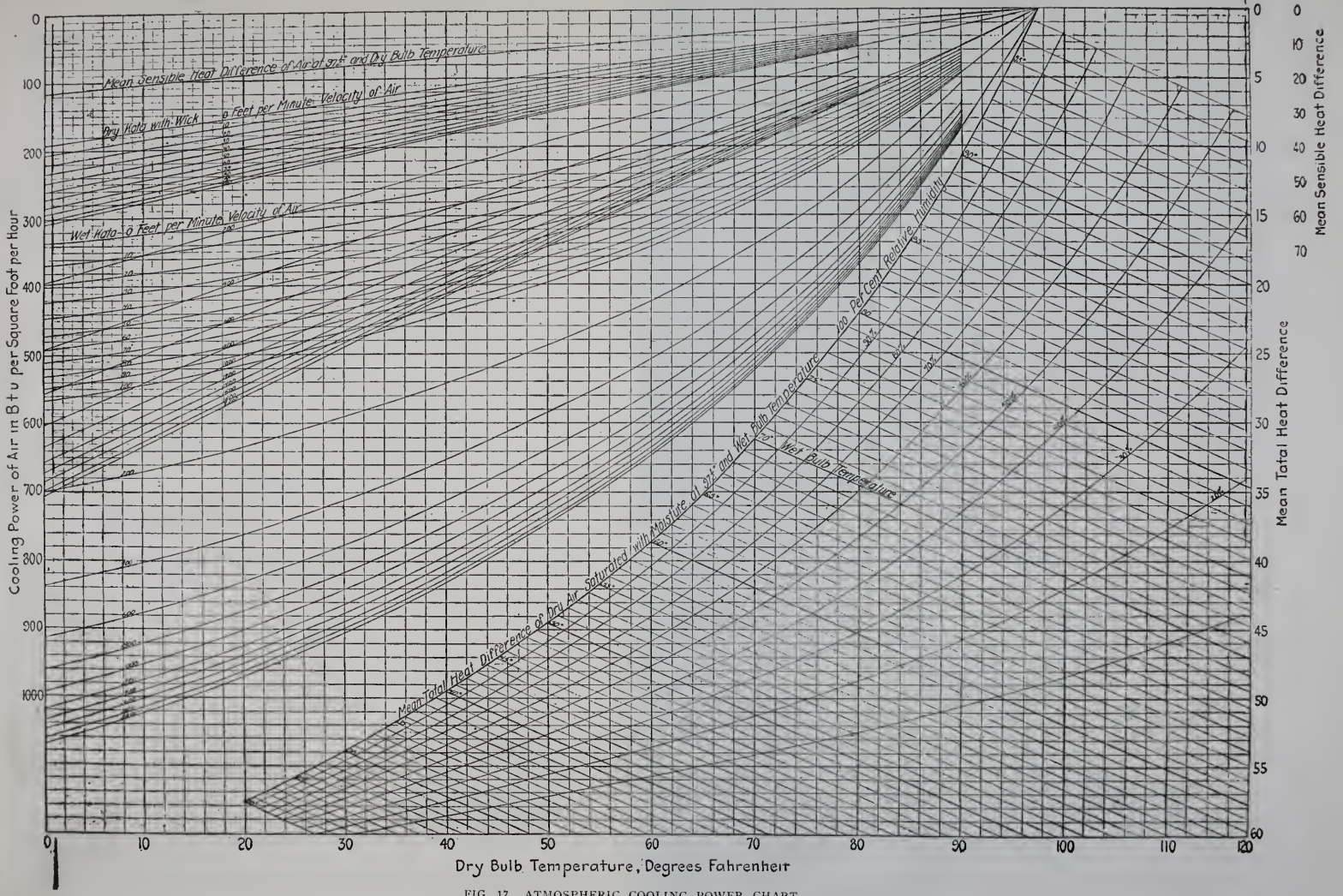
Mean Sensible Heat Difference of Air at 90° and Dry Bulb Temperature
Dry Kata with Wick
0 Feet per Minute Velocity of Air
Wet Kata 0 Feet per Minute Velocity of Air
Mean Total Heat Difference of Dry Air Saturated with Moisture at 92° and Wet Bulb Temperature
100 Per Cent Relative Humidity
Wet Bulb Temperature
Cooling Power of Air in B t u per Square Foot per Hour
Mean Total Heat Difference
Mean Sensible Heat Difference
Dry Bulb Temperature, Degrees Fahrenheit

FIG. 17. ATMOSPHERIC COOLING POWER CHART.

ward to the 263 B.t.u. line. The 80-ft. per min. line passing through this point of intersection is the required velocity.

Example (4). The wet and dry bulb temperature readings taken in a glass industry were 60 deg. and 80 deg. fahr. respectively in practically still air. A dry Kata with a factor equal to 0.01409 B.t.u. per sq. in. cooled in 365.2 seconds. To what extent are workmen affected by the heat radiated from high temperature furnaces? Referring to the chart it will be found that the dry cooling power for a dry bulb temperature

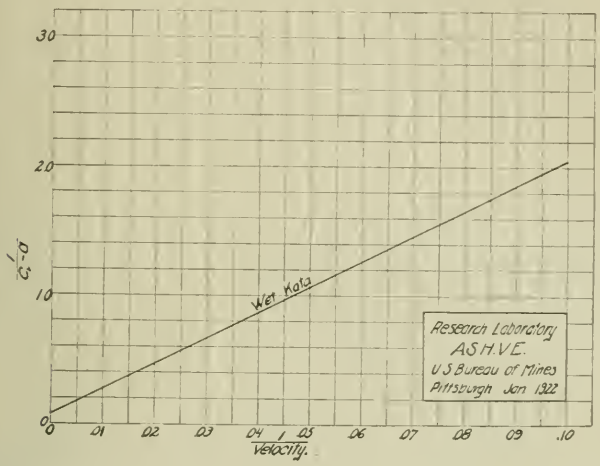

FIG. 16. DETERMINATION OF RELATION OF VELOCITY TO RATE OF HEAT LOSS.

of 80 deg. and still air is 35 B.t.u. per sq. ft. per hr. The actual cooling power of the air may be calculated by the formula

$$H = \frac{K}{\phi}$$

Substituting the conditions of the problem, then

$$H = \frac{0.01409}{365.2}$$
$$= 0.00003858 \text{ B.t.u. per sq. in. per sec.}$$

or $144 \times 3600 \times 0.00003858 = 20$ B.t.u. per sq. ft. per hr.

With no furnaces and with walls at the same temperature as the air, then neglecting other minor effects, the cooling power would be 35 instead of 20. The furnaces have therefore decreased the equivalent cooling power of the air 42.8 per cent.

In conclusion the writers desire to emphasize the necessity of using a

standard wick on all Kata thermometers and it is recommended that the cotton lisle wick be used for both the wet and dry readings.

The writers acknowledge their indebtedness to W. H. Carrier for many of the mathematical considerations of the investigation and for valuable suggestions given.

No. 621

OXIDES OF NITROGEN PRODUCED BY OZONE APPARATUS

By G. W. Jones[1] and W. P. Yant[1], Pittsburgh, Pa. (Non-Members)

and

O. W. Armspach, Pittsburgh, Pa. (Member)

INTRODUCTION

SINCE the time Schönbein, in the year 1840, announced the discovery of a new substance very active chemically and having a peculiar strong smelling odor, to the present time, ozone has been extensively investigated, not only from the standpoint of its chemical composition, chemical reactivity, detection, and estimation, but also its physiological, germicidal, bleaching, and deodorizing effects.

Many of these questions are still in the experimental state, for example: even the primary one of the exact chemical composition of ozone. Does the ozone molecule contain three, four, five or six atoms of oxygen, or a mixture of these, obeying certain chemical laws? But without going into detail with that reference to the work of past investigations, we may, at least for the present, make the general assumption that ozone is an allotropic form of oxygen in a more active and condensed state, consisting of three atoms per molecule, and all values in this report are based on this assumption. The questions that seem of greater importance, are those pertaining to the practical use of its properties, regardless of its structure, and first of all, suitable methods of analysis for the control of those investigations.

The question of ozonized air for ventilating purposes has received considerable attention during the last few years (and more so quite recently), both from the standpoint of engineers and chemists. Engineers see great possibilities, in the recirculation of the air used in the ventilation of public buildings provided that some suitable oxidizing agent as ozone can be used for destroying bacteria and odors, and some method used for removing the carbon dioxide produced; and only the admission of sufficient fresh air to keep the oxygen percentage nearly normal.

Ozone has been used as the purifying agent and good results have been reported. Likewise its use for purifying the air used in cold storage plants and similar places to prevent the spoiling of food products is

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[1] Assistant Gas Chemist, U. S. Bureau of Mines.

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Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, January, 1922.

receiving consideration. Also, it has been suggested for use in the bleaching of textiles and even for their sterilization. Should its properties be such that they can be used effectively without injury, there are possibilities that it will furnish a solution to many important problems.

Since these are all primarily ventilation problems, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS became interested in the subject of ozone for ventilation. On the other hand, the U. S. Bureau of Mines is also interested in ventilation, pertaining more especially to mining problems. As a result, a co-operative agreement was arranged between the Society and the Bureau of Mines, to make certain investigations pertaining to ozone, as outlined in the following program:

1. Quantitative determination of ozone and oxides of nitrogen in ozonized air;
   - *a.* Laboratory method.
   - *b.* Field method.
2. Concentration study of ozone and oxides of nitrogen;
   - *a.* Concentration of 0_3 producing a detectable odor,
   - *b.* Concentration of 0_3 necessary to destroy odors,
   - *c.* Concentration of 0_3 causing irritation,
   - *d.* Concentration of 0_3 necessary for germicidal effect.
3. Tests of ventilation systems using ozonized air to determine the effects of ozone in relation to factors in ventilation and the desirable concentration;
4. Use of ozone in connection with recirculation of air in buildings.

Several phases of this program are being carried out at other laboratories, and since it is not desirable to duplicate efforts, some of the parts given in the above outline may be eliminated.

It is of primary importance in an investigation of this kind to have suitable methods of analysis both for use in the laboratory and later in the field; consequently, the analytical methods were the first considered. For a laboratory method, that based on the oxidation of an iodine salt solution, such as potassium iodide and a subsequent titration of the iodine liberated with sodium thio-sulphate using starch indicator, proved suitable, as follows:

$$(1)\quad 2KI + O_3 + H_2O = 2KOH + O_2 + I_2$$
$$(2)\quad I_2 + 2Na_2S_2O_3 = 2NaI + Na_2S_4O_6$$

However, certain impurities such as chlorine or oxides of nitrogen will also liberate iodine from potassium iodide, and consequently would give erroneous results. In view of this, it was thought that if the potassium hydroxide formed by the ozone could be determined, along with the iodine liberated, a very good check could be secured, as indicated by reaction 1. With the low concentrations of ozone used in ventilation work, sufficient potassium hydroxide is not formed to permit a satisfactory determination by titration with a standard acid solution, or by a back titration with a standard alkali where the absorption is made in an acidified iodine salt solution, which is the better practice.

As a solution to this problem, the method used for the determination of ozone in this investigation was as follows:

The absorption is made in a 5 per cent solution of potassium iodide, made acid in slight excess to the estimated amount of KOH that will be formed; then the titration is made with hundredth normal sodium thio-sulphate (three thousandth normal can be used for work with very low concentrations), being careful not to overrun the starch end-point After the amount of thio-sulphate used is recorded, a few drops of a 25 per cent solution of pure potassium iodate are added and the titration with thio-sulphate continued for the iodine liberated from the potassium iodate by the excess acid. When the endpoint is again reached, the determination

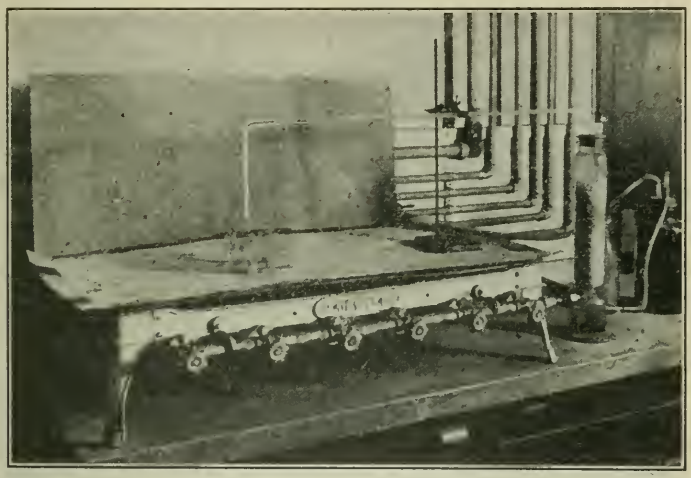

FIG. 1. APPARATUS USED FOR EXACT DETERMINATIONS OF OXIDES OF NITROGEN.

is complete, and the volumes of thio-sulphate used in the latter step should check the first, providing other acid oxidizing impurities are absent. Should some of the iodine liberated in the absorption tube be due to these impurities, less potassium hydroxide would be formed and a corresponding tendency toward an increased amount of acid, both of which would magnify the effect of the contaminating gas.

REACTIONS OF METHODS

(1) $2KI + O_3 + H_2O = 2KOH + O_2 + I_2$

(2) $2KOH + H_2SO_4$ (slight excess) $= K_2SO_4 + 2H_2O + H_2SO_4$

(3) $I_2 + 2Na_2S_2O_3 = 2NaI + Na_2S_4O_6$

(4) $3H_2SO_4 + KIO_3 + 5KI = 3K_2SO_4 + 3H_2O + 3I_2$

(5) $3I_2 + 6Na_2S_2O_3 = 6NaI + 3Na_2S_4O_6$

Granting that reaction 1 only goes 98 per cent to completeness, the error can be compensated for by standardizing the acid by using potassium

iodate and the thio-sulphate. Using aliquot parts of the solution through which ozone had been passed, the results indicated in Table 1 were obtained.

TABLE 1. OZONE PER CUBIC CENTIMETER

	From determination of I_2 liberated	From determination of KOH formed
Test (1)	6.88	6.80
	6.75	6.77
	6.81	6.65
Test (2)	8.11	7.97
	8.20	8.10
	8.20	8.00
Test (3)	0.842	0.857
	0.844	0.842
	0.839	0.836

FIG. 2. COMPARISONS OF SOLUTIONS WITH STANDARDS IN OZONE DETERMINATIONS.

OXIDES OF NITROGEN

As this investigation of ozone deals mainly with problems of ventilating public buildings, a second matter of importance would be a determination of other constituents that might be formed along with ozone and which would be deleterious to its use, both from physiological and material standpoints. Supposing that the conditions for the formation of ozone by an electrical discharge might also be somewhat conducive to the formation of oxides of nitrogen, this latter was the next part of the program considered.

Using the given method as a basis for the determination of ozone. different types of ozone apparatus were investigated to determine the amount of oxides of nitrogen produced and in what proportions in

[1] The Determination of Oxides of Nitrogen by Allison, V. C.; Parker, W. L., and Jones, G. W. Technical Paper 249, 1921.

[2] The Formation of Oxides of Nitrogen in the Slow Combustion and Explosion Methods in Gas Analysis, by Jones, G. W., and Parker, W. L. *Jour. Ind. & Eng. Chem.*, Vol. 13, 1921, p. 154.

comparison to the ozone. The method used was that developed by the Bureau of Mines for determining oxides of nitrogen from products of combustion of explosives[1] in mine air and gas mixtures[2] to which reference should be made for a detailed description.

Briefly, the method consists in absorbing the oxides of nitrogen in a 5 per cent solution of sodium hydroxide, adding H_2O_2, and carefully evaporating to dryness on a hot plate. A stream of air dried with sulphuric acid and cleaned with soda lime and potassium hydroxide is blown over the beakers to aid evaporation, and prevent contamination from the outside air, as shown in Fig. 1. When the precipitate is dry, 1 cc. of di-phenyl-sulphonic acid is added, the whole diluted with weak ammonia

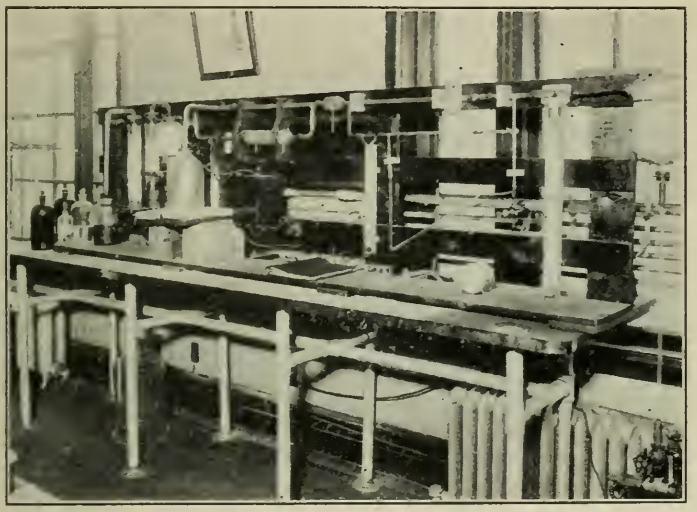

FIG. 3. APPARATUS USED FOR OBTAINING VARYING CONCENTRATIONS OF OZONE IN AIR.

solution, filtered into a Nessler tube, made alkaline with ammonia, diluted to 100 c.c., and compared with standards as shown in Fig. 2. The general procedure follows the usual practice for water analysis, but applied to gases.

In this paper, by oxides of nitrogen are meant only those oxides which are absorbed by the 5 per cent sodium hydroxide solution. This eliminates nitrous oxide NO_2 whose physiological effects are comparatively harmless in low concentration. It includes, however, nitric oxide, nitrogen peroxide, and other oxides as N_2O_5, and N_2O_3 if present. Nitric oxide in the presence of oxygen or air, is almost completely changed to nitrogen peroxide ($N_2O_4 - NO_2$), which in the presence of moisture is changed to nitrous and nitric acid.

APPARATUS

In carrying out the investigation, on analytical methods, an apparatus had been constructed as shown in Fig. 3, by means of which it was

possible to obtain a wide range of concentrations in the proximity of one part per million by a proper manipulation of flow meters, allowing part of the air to be ozonized and then diluted with air from a by-pass and the total flow again measured. Of this air-ozone mixture, a measured amount could be drawn through an absorption tube for analysis. To guard against impurities such as vapors and dust in the laboratory air, a cleaning and also drying train was provided through which the air to be ozonized could be passed when desired. In the construction of this apparatus, all joints exposed to ozone were made with a flange similar to that for sewer pipe connections, and the seal made with sealing wax.

As previously stated, this apparatus was designed primarily for work on analytical methods, but it was also found to be, after a slight modification, suitable for testing ozone generators for oxides of nitrogen. The

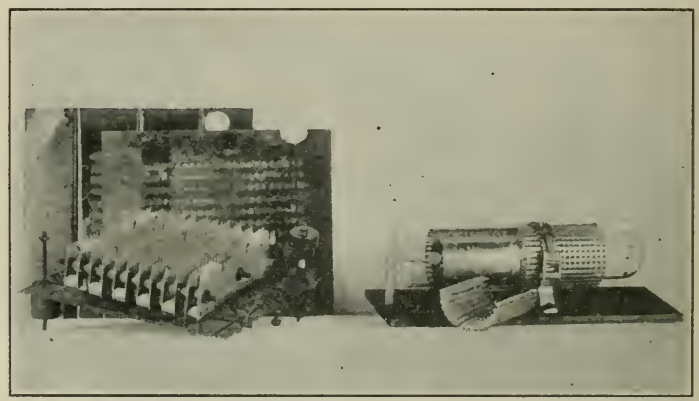

FIG. 4. TYPES OF OZONE GENERATORS USED IN INVESTIGATIONS.

modification referred to was necessitated by the different sizes and shapes of the generators. To accommodate all, a bell jar was connected between the cleaning train and the flow-meters (see Fig. 3), and the generators were placed therein, after which soda lime was banked around the outside, the air to be ozonized being drawn through between the bottom of the bell jar and the plate on which it was supported.

The two types of generators tested (see Fig. 4), were both commercial pieces of apparatus, designed for ventilation work. The square type is essentially a series of aluminum points protected from two iron plates by a sheet of mica. The tubular type was also a series of points stamped from a sheet of aluminum and which enclosed a pyrex glass tube, the latter being lined with a thin sheet of aluminum. The electrical discharge in both cases was caused to pass through the di-electrics and formed the so-called brush discharge.

Tests made for the formation of oxides of nitrogen gave the results recorded in Table 2.

From the above it is seen that the quantities of oxides of nitrogen

produced by the types of ozonators tested, are approximately one part in 150 parts of ozone. If one part of ozone per million is taken as the usual concentration for ventilation purposes, then the oxides of nitrogen concentration is only one part in 150 million, a quantity too dilute to produce physiological effects on the human body from our present knowledge.

TABLE 2. RATIO OF OZONE TO OXIDES OF NITROGEN PRODUCED BY OZONE GENERATORS

Rate of Flow, 840 c.c. per min.

Air flow c.c.	Ozone c.c.	Oxides N_2 c.c.	Concentrations Ozone Parts per Million	Ratio Ozone to Oxides of Nitrogen
		TUBULAR TYPE		
16800	6.88	.042	410	164-1
16800	6.88	.036	410	191-1
16800	7.88	.059	470	134-1
16800	7.88	.060	470	131-1
16800	8.11	.053	480	153-1
16800	8.11	.054	480	150-1
		SQUARE TYPE		
16800	7.13	.064	420	111-1
16800	6.44	.046	380	140-1
33600	14.56	.072	430	202-1
33600	15.15	.087	450	174-1
33600	12.30	.081	370	152-1
33600	14.70	.090	440	163-1
33600	14.70	.094	440	156-1

Data on just what concentration of oxides of nitrogen is necessary for physiological harm are by no means conclusive. Irvine[1] states that "nitrous acid fumes are intensely poisonous, and of all gases the most treacherous." "A certain number of cases returned as pneumonia, among underground workers, are really cases of nitrous acid fume poisoning, and the intensely irritating character of nitrous fumes has suggested that the repeated inhalation of small quantities of this gas may be maintaining a catarrhal condition of the air passages and lungs, and be a contributary factor in the development of 'miner's phthisis'." Lehman and Hasegawa[2] found that 39 (0.0039 per cent) parts per million can be tolerated by many individuals for several hours, and 78 (0.0078 per cent) parts can be borne for only one-half hour. From those investigators, it is seen that oxides of nitrogen exert marked physiological effects when the concentration is only 39 parts per million. When it is considered that many people may, where ozone apparatus are installed, work continuously in the presence of oxides of nitrogen, and also that its effects would have a tendency to be accumulative, the allowable concentration should be much

[1] Irvine, L. G. Accidents from poisonous asphyxiating gases in Mines. Coll. Guard, Vol. 111, 1916, pp. 653-655.

[2] Lehman, K. B., and Hasegawa, L. Studies on the action of technically and hygienically important gases and vapors upon man. The nitrous gases, nitric oxide, nitrogen peroxide, nitric acid. Archiv. fur Hygiene, Jahrg. 77, pp. 323-68.

lower than 39 parts as given above. What the allowable limit is, requires further investigation. At any rate, for the present, the quantities of oxides of nitrogen produced by the ozone generators tested, are so small, 0.007 parts per million, when the ozone concentration is one part per million, that the physiological effects from the oxides of nitrogen produced by these generators seem to be negligible.

No. 622

THE CONTROL OF BLOWER MOTORS

By Henry G. Issertell, New York, N. Y.
Member

THE control of motors for operating blowers and exhaust fans is so clearly related to the selection of the right motor characteristics for this service, that the two must be considered together. This paper is devoted to the selection of motors and their controls for driving enclosed fans of the paddle wheel, centrifugal, or multi-vane type for the ventilation of buildings. While the same characteristics apply more or less to other fan applications, such as disc or propeller fans, mine fans and positive pressure blowers, these applications are not here considered.

The basic conditions on which motors and controls are selected are as follows:

1. The characteristics of the current, whether alternating current or direct current.

2. The character of service, which involves the question of whether the fans are to run continuously or intermittently.

3. Whether it is possible to predetermine the horsepower requirements with accuracy.

4. Whether the requirements of air delivery involve a change in the quantity of air and the pressure from time to time.

5. The location of the fans and the personnel available for their operation.

6. The circumstances determining the running of the fan—that is, whether the operation of the fan depends on the judgment of the building superintendent for different weather conditions, use of the rooms, etc., or whether the fan is automatically controlled under predetermined conditions of temperature or pressure by means of thermostatic or pressure control independent of any manual direction.

These conditions involve again:

1. Constant-speed motors—manual-control, d.c. circuit.
2. Variable-speed motors—manual-control, d.c. circuit.
3. Constant-speed motors—remote-control, d.c. circuit.
4. Variable-speed motors—remote control, d.c. circuit.
5. Constant-speed motors—manual-control, a.c. circuit.
6. Variable-speed motors—manual-control, a.c. circuit.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1922.

7. Constant-speed motors—remote-control, a.c. circuit.
8. Variable-speed motors—remote-control, a.c. circuit.

REMOTE CONTROL

The remote-control equipment can each be subdivided into:

A. *Push-button control*, where the service switch is left closed and the motor started and stopped by push buttons from a remote point, under control of the building superintendent or other authorized person. This control may be either for one running speed (constant speed) or for any of several running speeds predetermined by the setting of controller handle, or dial-switch-arm as it is technically called (predetermined speed control), or any one of several speeds obtained from the push-button station (full-automatic control).

B. *Thermostatic control*, where an electric relay, responsive to temperature variations, acts as a predetermined minimum or maximum temperature to start or stop the fan motor through an automatic controller.

C. *Pressure control*, where a diaphragm or piston responsive to air pressure (or vacuum) acts through a relay to start or stop a fan motor. This type of control is more frequently used with pump motors than with fan motors, the pump starting and stopping with variations in open tank level or closed tank pressure. Pressure control is used with variable-speed, constant-running fans which are started and stopped manually to change the speed to meet varying air-delivery requirements, as for example, in fans supplying air for forced draft to furnaces.

D. *Line-switch control*, which is a form of control intermediate between *hand control* and *push-button control*. This may be applied to either a constant or variable speed motor which need be stopped or started from one point only. The controller functions to accelerate the motor automatically on closing the line switch. It is limited to small motors usually 5 hp. or less.

GENERAL CONDITIONS

The general conditions to be met by control apparatus applying to all types are under-voltage release, under-voltage protection, overload protection, and enclosure of live parts.

Under-voltage Release.—This means that, if the voltage fails, thereby shutting down the motor, the sudden restoration of voltage *shall cause no damage*. The motor may automatically start up again, but it must start up through its regular steps to insure keeping down the current inrush, and to prevent jarring the fan. A d.c. motor with remote control of the *line switch* type is thus protected. An a.c. motor that is normally started by being thrown across the line is obviously also self-protected. If there is no objection to a fan starting up automatically when voltage is restored after failure, under-voltage protection alone is sufficient on small motors.

Under-voltage Protection.—This means that if the voltage fails, thereby shutting down the motor, the motor can only be restarted by intentional action of someone operating the controller (if manual control) or pressing the start button (if remote push-button control). In the case of pressure or temperature control, shutting down the motor usually brings about a minimum condition which will cause the relay to act to start up the motor as soon as voltage is restored to the relay terminals.

In the case of controllers for variable speed, a.c., two- or three-phase, slip-ring motors, under-voltage protection is accomplished best by a mag-

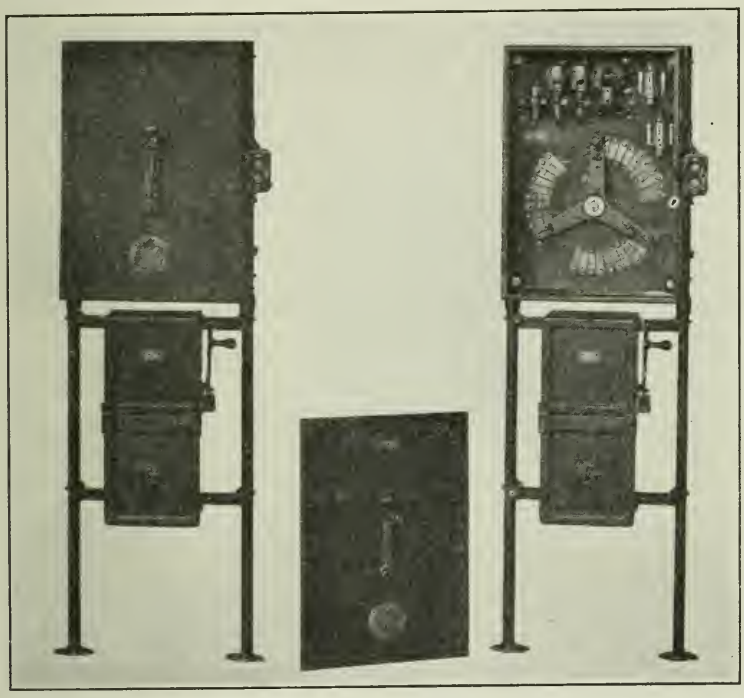

FIG. 1. ENCLOSED MANUALLY-OPERATED CONTROLLER FOR A. C. SLIP-RING TYPE BLOWER MOTOR FITTED WITH NO-VOLTAGE AND OVERLOAD PROTECTION.

netic main-line switch or contactor which is opened or closed by a push-button station. In such motors the speed control is obtained by resistance in the rotor or secondary circuit which is entirely independent of the primary circuit. It is this primary circuit which must be closed to start the motor, and opened to stop the motor. Therefore, fitting the speed-controlling handle with a spring return and holding coil as in a d.c. hand starter or controller, would require a multiplicity of extra and insulated contact segments, which would produce a very complicated and expensive control handle.

In Fig. 1 is illustrated an a.c. control for slip-ring motors with line contactor giving under-voltage protection. The push-button station out-

side the case has a start and a stop button. The controller handle must be all the way back to the left before the start button will function. This insures starting with all resistance in. The stop button is fitted with a pivoted clip which may be locked in place over the button if desired. When so locked the equipment cannot be started.

Overload Protection.—Underwriters' requirements in most sections of the country require insertion of fuses in the motor circuits or a circuit breaker, regardless of whether a separate overload relay is installed on the control panel. Hence the use of *overload release* is generally superfluous where overloads are of infrequent occurrence. In the case of hand-operated starters and speed regulators for d.c. circuits which have an under-voltage release magnet which holds the controller handle in place and lets it fly back to *off* position in voltage failure, the usual *overload release,* which operates to open or short circuit the under-voltage coil and so let the handle fly back to off position, is usually unreliable and of little value.

In the case of controllers fitted with a magnetic main-line switch, which is the case with the larger d.c. manually-controlled panels (above 50 h.p.), or with d.c., remote-controlled, push-button-operated panels, or with a.c. panels having low-voltage release, whether manual or remote control, a type of overload relay is available which is very reliable and can be set to close limits.

There exists considerable difference of opinion as to the relative merits of a fused switch and a circuit breaker. If no other overload protection device is installed, and if overloads are of frequent occurrence, the circuit breaker is better because it eliminates fuse renewals. As pointed out below, if overloads are of infrequent occurrence in fan application, and if a magnetic main line switch or contactor is part of the control, the addition of an overload relay, in connection with a simple fused switch of the enclosed type is quite as effective and less expensive to install than a circuit breaker of type that will really meet all requirements. On a. c. blower-motor panels especially, it makes a better combination.

Fan motors if properly selected are very seldom overloaded. If the calculations are incorrect and the duct resistance too high, there may be a permanent condition requiring more power, but the function of the overload relay is for protection against unusual overloads, which in the case of motor driven fans can come only from seized bearings, grounds in the windings and similar accidents which the fuses can take care of at very little expense because of infrequent occurrences.

There has recently been developed a *thermal relay* which functions in a most reliable and truly logical manner under overloads. The damage to a motor due to overload results chiefly from heating. The thermal relay is subjected to the same effects. Too frequent or too long continued heating cause the relay to operate and open the motor circuit through its magnetic contactor. Sudden overloads do not effect the relay. The more familiar time limit relay delays action on sudden brief overloads (which the motor readily handles), but its action does follow the effect of the overload on the motor as truly as does the thermal relay.

Enclosure of Live Parts.—A growing appreciation of the right of both the competent electrician and the unsuspecting layman to be protected

against accidental contacts with live parts of electrical apparatus, and reduction of fire risk, is bringing about a general adoption of totally-enclosed controllers.

The illustrations given show the evolution of the enclosed controller. Fig. 2 shows an open controller. The service switch is also exposed.

Fig. 3 shows a controller mounted with switch and fuses and all mounted in cabinet. This has been until recently looked upon as an ideal arrangement. However, when the door is open, the controller is alive and even when the switch is pulled, the incoming line terminals are alive.

Fig. 4 shows the ultimate in safety devices. The controller is in a separate cabinet with a locked cover. The service switch is in a separate cabinet so designed that its cover cannot be opened to remove fuses unless the switch is opened by moving the handle forward; and when the fuse

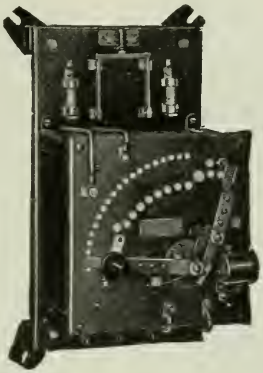

FIG. 2. OLD STYLE OPEN PANEL WITH CONTROLLER, SWITCH AND FUSES MOUNTED ALL TOGETHER

cover is open, the only live terminals are at the top where they cannot be reached without unscrewing the cover. When the service switch is open every part of the controller, including the incoming leads, is dead.

Enclosed controllers may be of two types: one in which the cover must be open to operate the controller handle, if it is a manual controller (Fig. 4) or to set the speed if it is a predetermined-speed remote controller, and the other type in which the controller handle is operable from the front of the case by an insulated handle. In the case of remote, predetermined-speed controllers, this externally-operated handle is simply a speed-setting device. The second type, with the insulated external handle is in a sense the safer of the two, because it is theoretically possible to open the cabinet door of the first type before opening the service switch. For control of fans in buildings, the former type is preferable, because once the cabinet door is closed and locked, the motor cannot be started and stopped or the speed changed by an unauthorized person, or a tenant of the building. This is especially important in schools. In industrial establishments, factories, etc., the former type with external handle is preferable.

The Electrical Safety Conference[1] has compiled a safety standard for

[1] Electrical Safety Conference, 25 City Hall Place New York, N. Y.

industrial control equipment which architects and engineers would do well to follow in their specifications and manufacturers in their designs. The organizations co-operating in this conference are the *Associated Manufacturers of Electrical Supplies, The Electric Power Club, The National Workman's Compensation Service Bureau,* the *Underwriters' Laboratories* and the *Bureau of Standards.*

SELECTION OF MOTORS

Next the selection of motors according to the eight main divisions given above will be considered. If direct current is available, d.c. motors will inevitably be used. If alternating current is available, a.c. motors may be used, or a motor generator or rotary converter installed and d.c. motors used. Frequently on a.c. circuits, both types are employed; d.c. motors

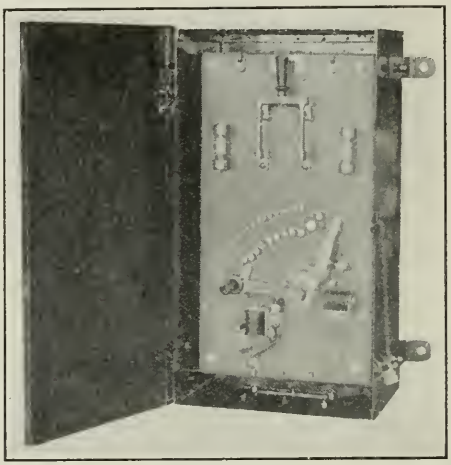

FIG. 3. PANEL WITH CONTROLLER SEGMENTS FLUSH MOUNTED, AND SWITCH AND FUSES ALL ENCLOSED IN CABINET.

on the variable speed blowers and a.c. motors on the constant speed blowers and on pumps.

Motors, either alternating current or direct current, may be of the open or semi-enclosed type. There is an increasing tendency towards the latter. Open motors are, however, entirely satisfactory in most cases and, located as they are in remote and seldom-reached portions of buildings and well protected from the weather, the fire or personal injury risk is very slight, and the extra cost of semi-enclosed motors, including the cost of the larger frames frequently needed because of the restricted circulation of air through the motors, is seldom justified. Totally-enclosed motors would seem never to be justified for such applications, though of course, in the case of disk fans where the motor is directly in the air stream, and hence subject to damage from foreign particles and moisture, enclosed motors are necessary. So are they in some pump applications.

The question as to the selection of 40 deg. and 50 deg. motors often

arises. There is a good deal of confusion existing in the minds of many engineers on this topic. So far as the ability of the motor windings and other parts to stand a temperature of 50 deg. cent. above the temperature of the surrounding atmosphere (usually taken at 25 deg. cent. or 77 deg. fahr.) as compared with 40 deg. cent. is concerned, that is for average cases an actual temperature of 75 deg. cent. (167 deg. fahr.) as compared with 65 deg. cent. (149 deg. fahr.), it may be said that 75 deg.

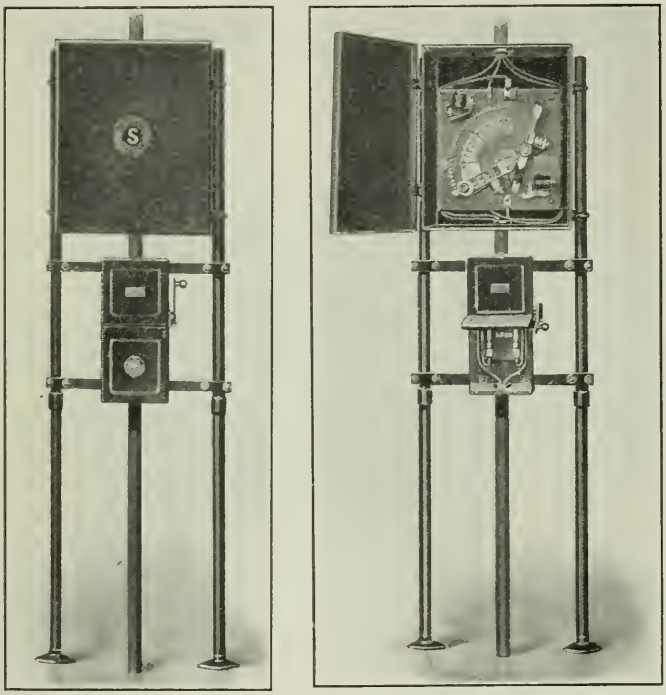

FIG. 4. THE LATEST DEVELOPMENT IN MANUAL CONTROL.

Enclosed D. C. speed regulator (all field control) with no-voltage and overload protection and with separately mounted safety-first type fused switch.

cent. is well below safe limits for ordinary insulation and therefore a 50 deg. motor is just as suitable for fan service as a 40 deg. motor and is usually less costly and more efficient. Especially is this true of slow speed motors, either direct current or alternating current.

This distinction is really one of overload capacity. Most commercial 40 deg. motors will stand a continuous overload of moderate amount in which case they virtually become 50 deg. motors. As stated above, fan motors are seldom overloaded unless the calculations go wrong. So far as the sudden or brief overloads are concerned, either type of motor will stand up if the controller protection functions properly. The selection of one type or the other is largely a question of prejudging the proper horsepower for the application.

HORSE POWER DETERMINATION

Now it is ordinarily customary to take the brake horsepower as given by the fan builders for the size fan and speed required for the given air delivery and duct resistance, this being a list value obtained by repeated trials, and add a safety factor for contingencies. This is usually 25 per cent. This factor in most cases is more than ample, especially as the figures are usually rounded out to the next highest. Thus assume the fan table gives 11 hp., adding 25 per cent gives 13.75 hp. and the engineer may call for a 15 hp. motor. To insist then on a 40 deg. motor virtually means a motor good for about 18 hp. continuously at 50 deg., or a safety factor of 63 per cent over the theoretical brake horsepower—surely this is a wide margin. Such a motor would be likely anyway to run a good part of the time at low speeds and greatly underloaded and hence inefficiently.

D. C. MOTORS

D.c. motors should be of the shunt type. If compound windings are used, they are for stabilizing purposes on a weakened field and the motor has shunt characteristics or fairly constant speed at different loads.

Whether d.c. motors for fans are of the commutating-pole type or not depends rather on the individual design than on the requirements of the application. The larger motors will be practically always of the commutating-pole type whether for constant or variable speed. The smaller motors may be of the non-commutating-pole type even for variable-speed service by field control if properly designed.

SPEED CONTROL

In the case of d.c., variable-speed motors, the method of varying the speed demands attention. The question of method involves the size of the motor, the amount of speed control desired, percent of time operated at low speeds, and the cost of power. The normal speed of a d.c. motor in the language of the designer is the speed for which it is designed to run under full load, without resistance in either armature or field circuit. The speed may be reduced below normal by resistance in the armature circuit down to ½ of normal speed, or it may be increased by inserting resistance in the field circuit. The amount of this increase varies, but for fan work the maximum is usually 1½ to 2 times the normal speed.

Fan manufacturers frequently refer to the normal speed as that at which the fan is usually operated. This may be the maximum fan speed or the minimum fan speed, or a point between. It may coincide with *normal* motor speed or it may not. It might cause less confusion, when speaking of fan speeds, to refer to the minimum, average and maximum speeds, and leave *normal* to the motor manufacturer.

When a motor speed is reduced by armature resistance, some power is wasted in the resistance and the motor consumes more power than is actually needed to run the fan and for the unavoidable or fixed motor losses. This fact has very properly led most engineers to specify field control on d.c. motors. But it is quite possible to overdo this.

The power required to drive a fan varies with the cube of the speed.

At ½ maximum speed, the power required is only ⅛ minimum. Therefore the power wasted in driving fans at low speeds on armature control is very little. For these reasons, the use of field control should never be carried beyond the point where the saving by elimination of the power wasted in the resistance is offset by reduced motor efficiency resulting from a larger motor frame size and by increased interest charges due to greater first cost.

There is also involved the first cost of the controller. This is greater for armature control or for combined armature and field control in the larger sizes than for straight field control, but the increase in motor cost for all field control is greater than the increase in controller cost for combined armature and field control.

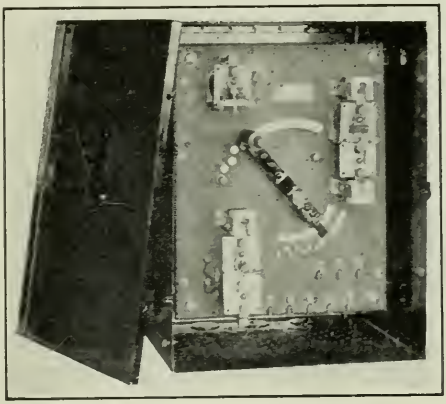

FIG. 5. D. C. REMOTE CONTROL SELF-STARTING SPEED REGULATOR FOR BLOWERS; COMBINED ARMATURE AND FIELD CONTROL

Automatic self-starting type for remote operation by push buttons. Controller arm can be set for any desired predetermined speed.

It is impossible to give rules which cover all cases but the following are representative of a number of conditions: under 5 hp. use all armature control unless cost of current is high (over 3½ cts. per kw.-hr.) and the motor is to operate practically all the time at low speeds; above 5 hp., use all field control if the total speed range is from a maximum fan speed down to 2/3 of maximum. (This means 50 per cent increase over motor *normal* speed by field). As the decrease from maximum diminishes, the field-control motor more closely approaches the armature-control motor in cost, and on the other hand, the loss of power in resistance if armature control is used diminishes.

If the conditions of the installation are so unusual as to require a total speed range of 2:1 or from maximum to ½ of maximum, the combination of field control and armature control, using 50 per cent field control (full speed down to 2/3 of full speed), and 25 per cent armature control (2/3 of full speed to ½ of full speed) will be found lower in first cost. This combination is to be preferred in place of the all field control whenever the cost of power is less than 3 cts. per kw.-hr., and where the

equipment will be operated at the lowest speed approximately 25 per cent of the time. An average case would show 25 per cent of the time operated at minimum speed; 75 per cent at 2/3 of maximum or above, that is, on field control with no power wasted. With such a combination, with power costing 3 cts. per kw.-hr., the combination of armature and field control will be decided by the lower first cost even if the motor is operated every day in the year. If the cost of power is less than 3 cts., or the motor is operated only occasionally or less than 25 per cent of the time on armature control, then the selection is all the more justified.

A. C. SUPPLY

Where a.c. supply is available, the question comes up as to whether to use a.c. motors or to put in a motor generator set and use d.c. motors. If the number of motors is small, or if the individual horsepowers are low, or if constant speed is acceptable, there is no question but that a.c. motors should be used.

In the case of larger buildings needing from 10 to 30 fans, taking from 10 to 50 hp. and requiring variable speed, the use of a motor generator set may well be considered. All elements have to be considered in each case; the overall cost, interest on the cost, overall efficiency, and the cost of power. The losses in a motor generator are from 20 to 30 per cent at average working conditions, and an equivalent saving by using d.c. motors with field control over a.c. motors with armature control must be realized to offset this. A rotary converter will usually be preferred to a motor generator set if it is possible to use alternating current for the lights, and constant-speed machinery. The somewhat decreased flexibility of voltage adjustment is offset by higher efficiency (losses 10 to 15 per cent instead of 20 to 30 per cent), as compared with a motor generator set.

The question of elevators enters into this problem as the d.c. supply may be desired for elevator operation in addition to variable speed blowers. A.c. motors, whether polyphase or single phase, are at a decided disadvantage compared with d.c. motors when it comes to direct connecting them to slow-speed blowers and exhausters. Such slow-speed motors have a very poor power factor, that is, a large *wattless* or non-power current, which means installation of larger transformers and cables and causes a serious disturbance to power companies' lines. There is a tendency for power companies to penalize low-power factor installations and this will tend to decrease rather than increase the cost of such motors.

If slow-speed motors are not used, speed reduction between motor shaft and fan shaft is obtained by belt, silent chain or gears. The belt drive is, of course, the cheapest but the least satisfactory. Chain drive is perhaps the best. Gear reductions using silent enclosed worm gears have found favor in some cases.

A.c. motors are normally rather noisy compared to d.c. motors, and special design must be resorted to if this noise is to be eliminated entirely. This noise, which is due to the alternating magnetic hum, is rendered unnoticeable in most cases by properly insulating the motor base and coupling against transmission of sound vibrations.

The so-called multi-speed, a.c. motor is frequently pointed to as a means of obtaining efficient operations without waste of power on a.c. circuits. The objection to such motors is the lack of flexibility.

Taking the case of a 60-cycle, belted or chain-driven motor, two speeds, 1200 and 600 r.p.m., can be obtained without very great increase in motor cost over a constant-speed motor. But these speeds are too far apart for blower work. 1200 and 900, or 900 and 720, or 720 and 600, can be obtained only by two independent windings, one of which is inactive while the other is working, thus increasing the motor frame to a great extent. Three speeds, 1200, 900 and 600 can be obtained with a motor rather larger than a slip-ring motor for the same job, and with a

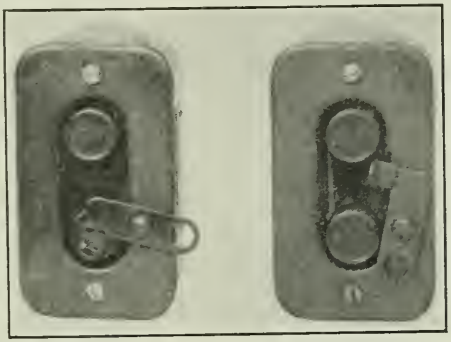

FIG. 6. PUSH BUTTON STATION FOR STARTING AND STOPPING

The motors are equipped with remote control. A latchet can be carried over the stop button and this latchet held in place with a padlock. When this lock is in place, the *feed circuit is* open and the equipment cannot be operated from this or any other station.

more complicated control. With slip-ring motors, or d.c. motors, just the right speeds can be had.

A type of motor which is rapidly coming into use is the commutator type, a.c., brush-shifting motor. In this motor the speed is varied by shifting the brushes. When the speed is reduced, there is no waste of current in resistances. This type of motor is therefore more efficient at low speeds than slip-ring motors, and is nearly as efficient as d.c. motors with field control. Any number of speeds can be obtained and the flexibility is therefore as great as a d.c. or a.c. slip-ring motor and much greater than the a.c. multi-speed motor.

In the smaller sizes, the speed is varied by manual operation, and this means by going to the motor to change the speed. For such applications, however, it is possible, unless extremes of speed change are needed, to leave the brushes set on a predetermined speed and start and stop the motor from a remote point just as with any of the remote-control devices described above.

With the larger motors, either hand control or automatic control is possible. With automatic control a small pilot motor is mounted on the motor and geared to the brush rigging. This pilot motor is controlled from a remote push-button station from which any speed can be obtained.

The brush-shifting motor is made in single-phase types up to 7½ hp.

and in polyphase type from 5 hp, and larger. The single-phase, brush-shifting motors, 3 hp. and smaller, are no more expensive than polyphase, slip-ring motors and are more efficient. The larger motors cost more than slip-ring motors of corresponding horsepower, but in many applications the increased carrying charges on the first cost will be more than offset by decreased cost of power, if the motor is operated any large part of the time at low speeds. Commutator motors are as a rule more

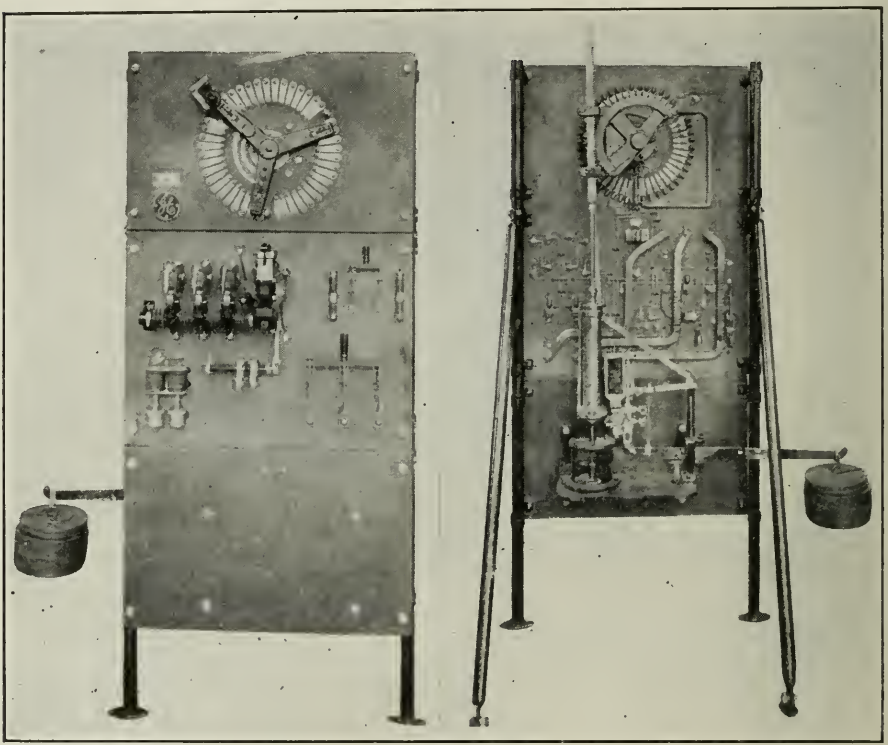

FIG. 7. AUTOMATIC SPEED CONTROLLER FOR A. C. SLIP RING MOTOR FOR FORCED DRAFT FANS CONTROLLED BY VACUUM REGULATOR

Through this regulator the speed is automatically adjusted to the requirements of the furnace as controlled by depression in the stack.

noisy than induction motors and this point must be considered in many building applications.

Remote control for variable-speed fan motors has many advantages over manual control and may in time come to be generally applied as the standard method, with manual control confined only to a few of the smaller and simpler installations.

A typical remote-control installation, whether for d.c. motors or a.c. slip-ring motors, will consist of an automatic, self-starting controller containing a handle or switch arm which can be set to any desired operating speed, the whole self-contained in a cabinet with cover which may be locked to prevent unauthorized persons from altering the speed.

Safety requirements will dictate the use of a separate, enclosed service switch and fuses. Fig. 5 shows such an equipment. A push-button station located at a suitable point gives control of starting and stopping. This may be in the building superintendent's office. In addition to this push-button station, additional stations can be used at other parts of the building if advisable. These additional stations may be both starting and stopping stations, or for stopping only. The push-button station may be fitted with a lock to prevent operation by anyone except those in possession of the key. See Fig. 6.

At a very slight additional cost, the remote control can be provided to give two different speeds from the push-button station: (1) a slow speed, the minimum; (2) a maximum speed which is the speed for which the dial-switch handle is set. This arrangement enables the superintendent to slow down the fan to a point where the air delivery will be at a minimum, but without entirely stopping the fan, and then speed it up again later as requirements may dictate. Then for a change in the regular high speed, a man is sent to the controller to change the setting of the predetermined speed arm. This may be required only occasionally with changes of weather conditions, etc.

At a somewhat greater cost, the control may be arranged to give several different speeds, 4 to 7, let us say, from the push-button stations and with an indicator to show the speed obtained. At a still greater cost, the control can be of the full-automatic type, in which the speed-controlling arm is operated by a pilot motor or magnets under control of the push buttons so that any speed or any change of speed within the motor range can be obtained. The chief value of full-automatic control is realized when the control is made responsible to changing conditions, that is, to temperature, pressure, etc. A special type of automatic control is shown in Fig. 7 for a forced-draft fan in which the speed is controlled from the vacuum in the back.

DISCUSSION

Carl F. Scott: I would like to call your attention to the work of the Electric Power Club on the standardization of control of this kind. The Electric Power Club is a group which includes practically all the manufacturers of electrical apparatus, and they have through their organization accomplished for the electrical industry a great deal of standardization, simplification, and reduction of types. Their work is perhaps most familiar by reason of their standardized specifications, under which electrical apparatus and supplies today are built, and these specifications, identical as they are, in most respects, with those of the *American Institute of Electrical Engineers,* constitute a protection to the buying public which has few equals in any line of mechanical endeavor.

The particular question of controllers for blowers is coming to the attention of one of the sub-committees of the Electric Power Club and it is believed that there will shortly be issued recommendations which the engineers can follow with full confidence in the same way that they can now issue specifications of generators or turbines.

I also want to comment on the use of the new type alternating current motors mentioned in the paper. I think they are going to be very largely used because of the fact that a reduction in speed can be accomplished

without the loss of power in resistance. That has been in the past one of the favorite arguments against the use of alternating current motors with variable speeds. While this new type is regarded as more or less in its infancy, it has been on the market for at least five years and a very large number of successful installations have been made for mine fans, industrial work, forced draft work and the like, with some very remarkable overall efficiencies as compared to the conventional slip-ring type motor. They are now mostly applicable to the larger sizes, but undoubtedly development and competition will bring them down to a point where they can be put into a building for fan service. At the present time they are more or less noisy as compared with other motors, but this can be overcome.

I was told the other day by one of the prominent engineers of the industry that alternating current elevators can now be constructed to run at 450 ft. per min. Only a few years ago 250 ft. was the limit. With an increase of speed to 450 ft. a min. it will make possible the use of the alternating current motor in the big buildings in the metropolitan districts, and with the cheaper cost of producing and distributing alternating current we may expect a wider use of alternating current in the big cities as well as in the outlying districts.

E. P. GOODWIN: What problems of commutation were obtained with the commutator motor that was discussed in the paper when shifting the brushes in various positions? It is well known that in direct current motors we have quite a problem. I wonder if such a problem exists in the alternating current type of motors.

CARL F. SCOTT: In regard to commutation of alternating current motors, that was the problem that probably held back the development of this type of motor for a good many years. Undoubtedly if the problem had not been a big one the motor would have been generally on the market ten or fifteen years ago. The problem has been successfully solved, however, by a scientific study of the reactions that go on within the motor, and the present day motors have been built in a way to largely reduce this commutation factor. It involves the proper proportioning of the winding on the armature, the use of a very large number of bars in the commutator, the use of narrow brushes, spanning only a very few bars, the under-cutting of the mica between the bars, and other less vital changes. The problem of commutation in direct current motors was solved fifteen or eighteen years ago by the commutating poles.

E. P. GOODWIN: May I ask further what is the efficiency of that motor in a particular size, say fifteen or twenty horse-power, in comparison with the direct current motor?

CARL F. SCOTT: The efficiency at full speed—I cannot give the exact figures for any particular motors, but this would be relative. If the efficiency at full speed of the slip ring motor was say 87 or 90 per cent, the efficiency of a commutator motor of the same type at the same speed would probably be two points lower; but when running at half speed, if the efficiency of the other motor was thirty-five per cent, we will say, counting the loss at the rheostat, the efficiency of the commutator motor would be around 65 per cent at that speed. So the value of the motor comes when there is any extensive amount of operation at low speed.

No. 623

DEHYDRATION AND FRESHENING OF CODFISH

By Henry W. Banks[1], 3d, New York, N. Y.
Non-Member

WHILE conditions for the vacuum dehydration of beef have been carefully determined by the work of Falk, Frankel[2], and McKee[3, 4], the literature appears to be lacking in data bearing on the treatment of other food materials by that method. The following study of the best conditions for the dehydration of codfish by the vacuum method was therefore undertaken. At the same time a study was made of the behavior of the dehydrated material upon freshening, or reabsorption of water. This fish is of particular interest in that it represents so large a proportion of the total fish supply, and may be considered to possess, in the main, the same characteristics as other lean salt water species. Very fatty fish, like salmon, present certain other qualities, from the point of view of dehydration, on account of their high content of low-melting fats. Such fish require a lower temperature for dehydration than the lean varieties, but present no other notable differences.

The experimental work upon which this discussion is based was embodied in an unpublished thesis by Joseph Karp, working under the direction of Professor Ralph H. McKee of the Department of Chemical Engineering, Columbia University, and it is through the kindness of the latter that this material was made available to the writer. For the sake of clearness and brevity the experimental results have been expressed graphically. Furthermore, the comparative behavior of fish dehydrated at various temperatures is thus brought out more clearly than would be the case if the results were expressed in tabular form.

The objects of the investigation were (1) to determine what temperature was most suitable for vacuum dehydration of the fish; (2) the rate of absorption of water, and quantity of water absorbed; (3) general considerations affecting the commercial value of the method.

In judging the quality of the finished product several points must be kept in mind. In the first place, the dehydrated fish, when allowed to absorb water, should resemble closely the fresh fish. It should be white in color and pleasing in appearance. The proteins should not be coagu-

[1] Banks and Craig, New York, N. Y.

[2] U. S. Patent No. 1,309,357, July, 1919.

[3] Falk, K. G., Frankel, E. M. and McKee, R. H., *Journal of Industrial and Engineering Chemistry*, Vol. 11, 1919, p. 1036.

[4] McKee, Ralph H., Journal of the American Society of Heating and Ventilating Engineers, March, 1920, p. 155.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1922.

lated. It should not present the appearance of having been cooked. In other words it should be, so far as possible, fresh fish with the water removed to an extent sufficient to prevent bacterial action and the formation of molds. When fish is dried by methods which include salting, pickling or smoking, its flavor and other characteristics are considerably altered. This is not the case when a properly controlled system of dehydration is employed. In this regard, it is advisable that the removal of water should be carried on at the lowest temperature economically possible, but high enough so that the process shall not be unnecessarily prolonged.

The apparatus used in this work was a vacuum shelf dryer with an effective drying surface of 75 sq. ft., equipped with a surface condenser and dry vacuum pump. A vacuum of 27.5 in. was employed throughout. The codfish to be dehydrated was cut into slices approximately ½ in. in thickness, and placed upon woven wire screens which rested upon the heated shelves of the dryer. Steam at sub-atmospheric pressure was employed as the heating medium in the shelves, though in commercial practise hot water may frequently be more convenient. 55 deg. cent. (131 deg. fahr.) was the lowest temperature used. The fish was periodically removed from the dryer for weighing.

In determining the time the fish required to re-absorb water, the slices were placed in water at room temperature from which they were removed at intervals, dried superficially, and weighed. The accompanying curves express the results so obtained, in which ordinates are percentage weight of fish referred to original sample of fresh fish, and abscissæ, the time.

The samples prepared at 55, 60, and 65 deg. cent. (131, 140 and 149 deg. fahr.) were very satisfactory in appearance, the last showing a slightly brown coloration. This color was somewhat deeper in the 70 and 75 deg. cent. samples (158 and 167 deg. fahr.), and the 80 deg. cent. (176 deg. fahr.) sample was dark brown and possessed a distinct odor of having been cooked. From the appearance alone it may be said that *cooking* takes place at temperatures above 75 deg. cent. (167 deg. fahr.). There is certainly partial coagulation of the proteins even below that temperature.

It will be noted that the samples dried at 55, 60 and 65 deg. cent. were somewhat irregular, but that all samples had lost approximately 65 per cent of the original weight after 9½ hours dehydration. Over a portion of the curve the 60 deg. cent. sample appears to be losing water more rapidly than the 65 deg. cent. sample. Periodical weighing of the material, which necessitates breaking the vacuum and removal of the samples with consequent cooling, would largely account for this irregularity and the unusually long total period of drying. Another source of error due to working with small samples may be referred to. It must be clear that the efficiency of the method depends largely on rapid heat transfer from the shelf to the material. With the samples of 2 or 3 lb. used in this work, a difference in physical contact between the pieces of fish and the wire tray, or between the wire tray and the iron shelf, would introduce noticeable variations in the rate of drying. Such variations would be largely eliminated by employing 100 lb. or more of fish for each run. This, however, is a minor point compared with the effect of temperature

of drying upon the final product. It is almost unnecessary to emphasize the fact that, as the drying is due to conducted heat, the tray must make accurate metallic contact over the surface of the shelf to insure an even removal of water from the material.

The rate of drying of the samples treated at 70, 75, and 80 deg. cent. was extremely regular, in spite of the fact that they were removed from the dryer as often as were those dried at lower temperatures. However, the samples dried at 70 deg. cent. and above lost water very rapidly during the first two or three hours, after which period the first weighing was made. It is clear that a constant rate of drying will not be reached until the material has become warmed throughout to nearly the temperature of the shelves, and this condition is more nearly attained when

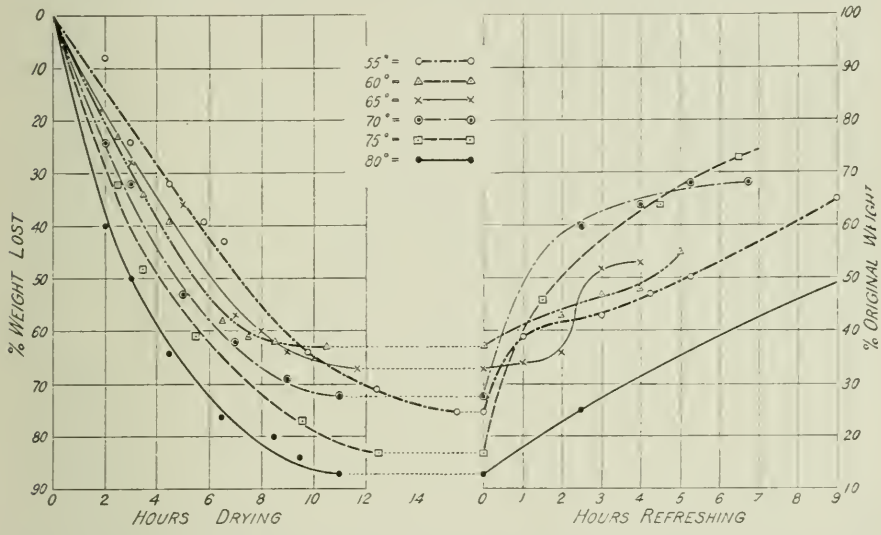

FIG. 1. CURVES SHOWING RATE OF DRYING AND WATER ABSORPTION AT DIFFERENT TEMPERATURES

considerable water has been lost and the cooling effect of evaporation becomes less. In practice, where the material is not removed from time to time for weighing, the rate of drying will be more regular and the total time of drying shorter than is shown by the accompanying curves About eight hours may be taken as the average drying time for codfish; in practice this means a reduction in weight of about 65 per cent. *Conditioning* or storage in fairly dry air for two or three days before packing will result in a further reduction of weight and equalization of the water content of the mass. Codfish reduced to 28 to 30 per cent of its original weight will apparently keep fresh indefinitely.

The rate of water absorption was somewhat irregular in the case of those samples dried at 55, 60 and 65 deg. cent. In no case did the sample regain entirely its original water content. A further factor in this connection may be noted. It is probable that these samples of fish lost a portion of their water-soluble proteins during the soaking process, a

fact which is indicated by the irregularity in the increase in weight, and by the greater apparent absorption of those dried at 70 and 75 deg. cent. It is safe to assume that the proteins of the latter were rather completely coagulated, while the sample dried at 80 deg. cent. showed such a marked change in color and odor as to give evidence of a much more deep-seated change. Its rate of absorption was very much slower than that of any of the other samples, due to the hardening of the protein, rendered tough and insoluble by the temperature employed. The drying curves give still another proof that a decided chemical change has occurred at this temperature. The other curves (omitting the 75 deg. cent. curve) show a minimum at about 75 per cent water, at which point constant weight is reached. This may be called the percent of uncombined water in this particular fish. The sample dehydrated at 75 deg. cent. has lost 84 per cent of its weight, and the 80 deg. cent. sample, 87 per cent of its weight, another fact which points clearly to the same phenomenon, a chemical decomposition which begins to occur between 70 and 75 deg. cent. Empirically, we should say that the 75 deg. cent. sample had begun to be cooked, and that the 80 deg. cent. sample had been cooked rather completely. Thus, loss in weight, rate of absorption of water, and physical appearance, taste, and odor agree quite definitely. Roughly, then, we may divide the samples into several classes: 55, 60 and 65 deg. cent., water removed without decomposition or coagulation of the protein; 70 and 75 deg. cent., coagulation fairly complete and chemical change becoming apparent; 80 deg. cent., chemical change very apparent, marked change in color, protein tough and slow to absorb water.

As stated previously, the most satisfactory method of judging the final product is almost entirely an empirical one. The determination of water-soluble protein, the amount of water absorbed in a given time, etc., while interesting, do not yield information of as great value as does a careful examination of physical state, odor, appearance, and flavor. The sum of all evidence now at hand indicates that 60 deg. cent (140 deg. fahr.) produces the best results. While 65 deg. cent. appears from the above data to be not too high a temperature, in practice we have found that the slightly darker shade of dehydrated fish thus produced is not quite so satisfactory as the 60 deg. cent. product. The latter is white and pleasing in appearance. In commercial practice, 60 deg. cent. $\pm$ 3 deg. cent. may be considered the appropriate limits. At very much lower temperatures bacterial action will exert a considerable influence, being favored by the temperature, and by the length of time required to produce the desired removal of water. The effects of higher temperatures have been referred to at some length above. It may be further noted that changes due to oxidation are virtually absent at the degree of vacuum employed.

While the above data refer to cod, it may be mentioned that the following fish have been successfully dehydrated by the vacuum method: flounder, sole, scallops, oysters, clams, soft shell crabs, crab meat, lobster meat, perch, butterfish, salmon, and shrimp.

In closing, a word as to the most practical method of cooking dehydrated fish is perhaps appropriate. It will be clear from the preceding discussion that the fish should be soaked in the smallest possible amount of water, and should be cooked in the same water so as not to lose

extracted soluble proteins and other extractive substances. Its use boiled, baked, or stewed is for that reason to be preferred to broiling or frying, as the latter methods necessitate discarding the water used to refresh the fish before cooking. It is very possible that for certain uses fish dehydrated at temperatures sufficiently high to render the proteins insoluble might be preferable, in that less material is lost on soaking in water. In general, however, the end desired is the removal of water, with the minimum change in the substance dehydrated, and the present usage of the term dehydration implies that such is the case.

The writer wishes to express his thanks to Professor McKee for placing at his disposal the experimental data on which the curves are based, and to Dr. K. G. Falk of the Harriman Research Laboratory for kind and helpful advice.

No. 624

HEATING THE AVERAGE HOME ECONOMICALLY

By P. J. Dougherty, Glendale, Calif.
Member

IN magnitude and importance this problem is second to none which the heating engineer is called upon to solve. In considering the economical operation of a new plant or how to improve that of an old one, there are nine principal factors or conditions that must be co-ordinated and examined closely. If any one of these nine conditions is below par, it will result in an unsatisfactory heating plant that otherwise would be a great success.

The owner of the house is responsible for four of those nine principal conditions, the heating contractor for three of them and the heater manufacturer for two of them. Due to the absence of recognized standards and fixed responsibilities laid down by a competent, impartial authority, such as the American Society of Heating and Ventilating Engineers, one of the favorite indoor sports in winter time amongst those three parties in case of heating troubles is *passing the buck* from one to the other.

THE OWNER

The *owner* is responsible for and should furnish a properly constructed building, ample draft, suitable fuel and intelligent operation of the heating plant. Any competent heating engineer can furnish a good heating plant, but the owner himself must furnish the brains to operate the plant properly if he expects economical results. A Government bulletin on saving fuel in heating boilers[1], issued during the war, states: "Unquestionably, with any fuel, the prime factor determining fuel consumption and freedom from operating troubles, although it may not be generally recognized, is *method of operation.*"

In a survey made during the war, it was discovered that practically as much coal was required to heat a house in Texas as one of the same size in Minnesota, in spite of the great difference in the average winter temperatures. In the cold states houses are well built and thus require a relatively small amount of fuel to keep them warm, as compared with the poorly constructed houses in the warmer states. Heavy building paper and shingles placed over the siding of a poorly constructed frame building will materially reduce coal bills besides reducing painting bills. Metal

[1] Technical Paper No. 97, U. S. Bureau of Mines, Saving Fuel in Heating a House, by L. P. Breckenridge and S. B. Flagg.

Paper presented at the Annual Meeting of the American Society of Heating and Ventilating Engineers, New York, N. Y., January, 1922.

weather strips on all outside doors and windows are very essential for economical heating. The leakage around three average sized windows in a room is frequently equal in area to a pipe 8 in. in diameter.

A strong draft, properly controlled, is the best fuel economizer on the market. Faulty chimneys are responsible for more heating troubles than any other condition. A highly efficient heater will prove a failure on a poor draft. A chimney less than 35 ft. high is erratic and uncertain in its action, and usually gives trouble in rainy or snowy weather. A tile 12 in. x 12 in. is the best size for the average house. No tile less than 8 in. x 12 in. should be used. A 4 in. tight brick partition should separate all tile lined flues in a chimney. All heater flues should be subjected to a smoke test and made tight before being accepted. Leaky flues are the most frequent cause of draft troubles. No other connection should be made to the heater flue but that of the heater.

No fuel should contain over 10 per cent ash. For every increase of 1 per cent in ash over this amount, the coal bill is increased 2 per cent. If the coal contains 25 per cent ash, 30 per cent more coal will be burned than if it contained only 10 per cent ash, as stove size contained before the war. The smaller sizes, such as pea and buckwheat usually contain 20 to 30 per cent ash. None of the slate and rock is removed from those smaller sizes at the coal breakers, as it is from the larger domestic sizes by the automatic slate pickers. Stove or range is the best size to use in most heaters. Egg size is too large as a rule; chestnut is best for small tank heaters; buckwheat and pea usually contain too much slate to be used economically for heating, except for banking, but with coke, pea and chestnut sizes are the best. Large sizes of coke should not be used.

Next to a good draft, a good fireman is the most important factor in fuel economy. The time to save fuel is in the fall and spring, and not in midwinter. Most people overshake their fires in fall and spring, and do not remove enough ashes from the firebox in midwinter. The main object in shaking the grates is not to remove ashes from the fire, but to make room for enough coal to carry till the next firing period. Do you get that important distinction? Shake the grates in fall and spring only when it is necessary to make enough room for the amount of coal necessary to carry till the next firing time. At all times in mild and cold weather, carry a deep fire up above the bottom of the fire door, or about 16 to 20 in. over the grate. A shallow fire invariably means high coal bills, frequent attention and unsatisfactory heat. The milder the weather the deeper the ash bed on top of the grates and the thinner the hot coal bed. In mild weather there may be only 6 or 8 in. of live fire and 12 in. of ashes over the grates. In real cold weather, there may be only 2 in. of ashes on the grates and 16 in. of live fire. When a fire is run properly this relation of live fire to dead fire over the grates will change with the weather, although the depth of both, from 16 to 20 in. should always be the same after firing. In shovelling fresh coal on the fire never entirely cover up the red coal or smother the flame, except when banking. When placing the first charge of coal on the fire, bank it up as high as possible against the back of the firebox and let it taper down to a feather edge in front so as to be able to see the fire shine or red coal near the fire door. All drafts should be turned on after the first coal charge.

In about 10 to 20 minutes blue flame shows all over the fire, after which the second and usually last charge of fresh coal should be fired. After the heater reaches the desired pressure or temperature the drafts should be properly adjusted. Proper draft control is a big factor in fuel economy.

Both heater and piping should be covered with asbestos. It is a great mistake to place a coil for heating domestic water in the firebox of any heater, unless the coil is above the lower edge of the fire door so as not to be down in the fire. A hot water supply boiler for such purposes is the most economical. While airing a room shut off the heat to that room, and if hot water heat, throw a cover over the radiator.

THE HEATING CONTRACTOR

The *heating contractor* and engineer are responsible for the proper capacity and proper construction of the heating plant and the type of heater used. Unfair competition and *peddling bids* on the part of the owner sometimes force a heating contractor to install a heating plant considerably below the capacity required. When properly operated and controlled, a plant that is above the capacity required is far more economical and satisfactory than one that is below the capacity required.

Most of the heating boilers installed in the average home are too small to meet the practical operating conditions demanded of them. The average man who attends his own heater is away from home about 10 hours during the day. If he uses hard coal, his heater, if of the proper size, should keep his home warm from the time he leaves in the morning until 10 hours afterward when he returns at night without his family being compelled to put coal on or shake the fire during the day.

It is the general practice to install boilers in homes rated with hard coal on only an 8-hour firing basis instead of on the 10 or 12-hour rating basis as the actual conditions require. Since the owner knows little or nothing about ratings, the *Heating and Piping Contractors Association* and the Boiler Testing Code of the Society should put an end to this bad practice by insisting that boilers for residence work be installed on at least their 10-hour rating basis. Furnaces as a rule are rated on a 10-hour firing basis.

The ratings placed on boilers tested according to the Boiler Testing Code of the Society, apply only to standard testing conditions such as ample draft, intelligent firing, good quality of coal, clean boiler properly cemented and mounted, etc. A boiler properly rated at say 800 ft., when tested under standard operating conditions, may not be able to develop say 500 ft. rating if the draft is weak, the firing inefficient, the coal of poor quality, the heating surface covered with soot, the sections and door frames leaking air or if there is short circuiting of the gases due to faulty cementing, etc. Because of those prevailing handicaps placed on most heaters, in which the actual operating conditions are far below standard testing conditions, good practice demands that small boilers be installed from 50 to 100 per cent greater in capacity than the amount of radiating surface installed. Good engineering practice demands that proper factors of safety be used to meet the varying conditions met in actual operation.

The short circuiting of the hot gases through a boiler and the leakage of air between the sections into the flue travel because of improper cementing of the spaces between the sections is a frequent cause of heating trouble. We all know that air leaks through cracks in a cigar's wrapper ruin a cigar. Air leaks because of faulty cementing also ruin the economical operation of a boiler. Regular boiler cement should be used; asbestos is porous and should never be used.

It is important that the boiler be the proper type to suit the conditions. If the draft conditions look unfavorable, install one size larger firepot with fewer intermediate sections having a short fire travel. If the draft is strong, give preference to a long fire travel. A heating plant that is improperly designed and poorly constructed is bound to be wasteful in fuel.

THE MANUFACTURER

The *heater manufacturer* is responsible for the proper rating and design of the heater. The Boiler Testing Code of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS sounded the death knell to the old *swivel chair* method of rating boilers, when the sales managers and officers of a company would gather around the directors' table, on which were placed the catalogs of their leading competitors. After finding out what ratings their competitors gave to boilers of certain size grate areas, it was merely a question of assigning as high a rating as the traffic would bear. The part of the Code covering rating is the real heart of it. It is the only part of the Code that is of any practical value to the average heating contractor, the designing engineer and architect. This Code began to take form at the January, 1914, meeting of the Society. The report of the Code Committee clearly sets forth that the specific object of the Boiler Testing Code shall be "the determination of the amount of steam which a boiler will furnish when burning a definite quantity of coal per hour for a given number of hours without attention to the fire", taking into consideration only those items of the test "which are essential in the proper rating of the boiler for commercial or sales purposes" (TRANSACTIONS, Vol. XX, 1914, p. 198). The two formulæ set forth in this 1914 code report for boiler capacity and boiler rating mark the real beginning of establishing a uniform standard for the commercial rating of low-pressure boilers. Unfortunately, the later revisions of the Code merged both of those formulæ into one.

It is only when burning hard coal in a boiler that a rating can be assigned to it, which in practice ranges from 3-hour to 12-hour ratings. In burning soft coal in a boiler, a certain capacity only can be assigned to the boiler and not a rating, which means developing a certain capacity per hour for a specified number of hours without attention of any kind to the fire. It is to be hoped that the next revision of our Low-Pressure Boiler Testing Code will clearly set forth the proper distinctions between hard-coal ratings and soft-coal capacities. The section on rating is the teeth of the code.

The design of a heater is a basic factor in determining the economy of fuel, maintenance and attention. Every steam boiler has three functions to perform, viz.: heat generation from the fuel; steam generation

from the water; and steam separation from the water in the steam dome. A deep fuel bed and provision for mixing the gases are the most important factors in developing heat from the fuel. The absence of soot pockets, easy access for keeping the heating surfaces clean, proper impingement of the hot gases and all heating surfaces backed up by a positive, rapid circulation of water are the major factors in determining the efficiency of the heating surface in developing steam. Steam may be separated from the water in the steam dome by gravity or by suitable baffling. Keeping a heater clean is just as important as keeping a machine oiled.

It is to be hoped that the Society will establish proper standards and define the various responsibilities for economical house heating and that the *Heating and Piping Contractors National Association* will see that they are enforced.

DISCUSSION

E. H. Lockwood (written). This paper is full of valuable suggestions on heating matters, where good advice is much needed. In my opinion the paragraph on handling the fire, beginning "Next to a good draft," is a model statement of how to handle a furnace efficiently, and deserves wide circulation. The paragraph on choice of fuel seems to me less satisfactory, because of its initial assumption of ten hours heating on one charge of fuel. Of course this demands a large size furnace and large size fuel.

The average householder has to manage a smaller unit, with a shorter firing period, say four or five hours. His chance to promote economy lies in the *choice of fuel.* I know persons who have used no anthracite coal larger than washed pea for many years, and claim that they use but little more weight of this fuel than the larger size. Others have used a combination of buckwheat with gas coke, with marked saving in cost. This combination is an ideal one where coke can be obtained, as it permits plenty of heat when desired, with excellent control of the fire by addition of the finer fuel. The cheaper fuels require more attention in handling the furnace, but in many cases this additional attention can be given by some member of the family.

The magazine heater, using buckwheat coal, has many good points for heating the home economically, and with the minimum of attention.

There is one factor in fuel combustion in residence heaters that is rarely mentioned in these discussions, namely, appliances for the admission of air above the fuel bed. The manufacturer, as a rule, provides only the adjustable air grids in the fire doors. These air openings are very unsatisfactory for proper admission of air, as is proved by the general practice of running the furnace with the openings closed all the time. Various appliances have been devised to improve this secondary air admission. Often the devices have been marketed with extravagant claims for fuel saving. I am satisfied that many of these devices have real merit, notwithstanding the *patent medicine* methods used in advertising them.

I have experimented for several years with two of these patented ap-

pliances. One consisted of a casting secured about ten inches above the normal fuel bed, near the center of the fire pot, drawing its air through a ¾-in. pipe fastened to the fire door casing. The other consisted of a special casting located in the smoke pipe close to the furnace, with small air openings that admitted air to the smoke pipe. At first thought it seems improbable that air thus introduced could return to the flues of the boiler and aid combustion of the burning gases. There is reason to believe that this actually happens, and furthermore, its effect is most marked when the draft is checked and the fire is burning slowly.

My experience with both of these appliances was most satisfactory. The benefit of the air admission was noticeable when the draft was checked off. Under these conditions, the fuel burned just as slowly as before, but gave out more heat to the furnace. This was especially true of the smoke pipe air admission. To make sure that my imagination was not affecting the results, I have repeatedly sealed up the opening for the air admission, thus rendering the device inoperative. In this case no difference was observed when the drafts were on and the fire was burning hot, but a marked difference was noticed when the draft was checked. I have no means of measuring the saving in fuel produced by either of these devices, other than by estimate. I am convinced, however, that a material saving of fuel can be effected in this way, and that manufacturers of boilers and furnaces should give more attention to proper admission of air above the fire.

J. R. McCOLL. I do not see where Mr. Dougherty has brought in the engineer's responsibility.

THE AUTHOR. The engineer will come in as the owner. The owner of the property is always responsible for the acts of his agent, and the engineer may be considered employed by the owner. This talk was entirely centered upon the little average home, that is usually built without an architect on the job, and very seldom with a consulting engineer. It is up to the boiler manufacturer to be responsible for the capacity factor to the heating contractor. But, however, the heating contractor must assume the responsibility for the construction of the heating plant, irrespective of design, and then a suitable boiler for the job, which responsibility also would devolve upon the designing engineer. Also the architect would be responsible for the construction of the chimney; that is, unless other considerations prevail. Of course the architect is always responsible for a poorly constructed building.

CHARLES A. FULLER. I see the author has some objection to hollow tile on the inside wall. I should like to know if he has the same objection to hollow tile on the outside wall.

THE AUTHOR. I have had some very sad experience with hollow tile this last fifteen years and my principal objection is this. In hollow tile construction with butt joints the webs are approximately about five-eighths of an inch thick and when they butt up against the other web I maintain it is a physical impossibility to maintain mortar joints tight enough to stop infiltration. There is another very serious objection. I know at our own plant we built a garage very recently and built it of

hollow tile and there was not a square foot of that job I couldn't see daylight through when it was completed. And then of course we know that the mortar or stucco outside does not prevent leakage unless a special treatment is employed such as building paper that is impervious to water or air, or some wash of some kind that is of an oily, tarry nature. I always increase my radiation 30 per cent where hollow tile is used, and have been on some jobs where it had to be done after the work was installed on account of the tremendous infiltration, or air leakage. In a building where they put the brick on edge, which is a criminal way, you have a tremendous induced current within the hollow wall and very great infiltration from it.

A Member. Mr. Dougherty has laid stress on draft. I would like to ask him if he does not find in the last two or three years the chimneys are built much better. We find that they are improving all over, and that improvement has come from the action of the engineers and architects.

The Author. I do not think there is a question about that. Every boiler manufacturer has a boiler to fit every flue, so if he runs into trouble with one boiler he merely takes out the pancakes and furnishes a boiler that will suit the condition, but it is expensive at the fuel pile. The recent chimney ordinance submitted by the fire underwriters is a tremendous step in the right direction and I look for that doing a great deal of good in the future to help the boiler manufacturer to market a much more efficient boiler than he is compelled to market today.

A Member. I think we are losing sight of a very important thing in this. I have had about seven or eight years' experience in heating and ventilating. I live in a house now heated with a warm air furnace. It is a detached house, with shingle weatherboarding and papered. On the roof there is nothing but shingles. I do not care anything about the infiltration. It is a small matter compared with the amount of air that leaks out through the shingles on the roof. In the majority of houses in Philadelphia with roofs covered with shingles, after a slight snow, the snow which projects over the shingle itself is entirely gone and that which remains over the eaves or cove of the roof stays there till the sun melts it off.

I have been experimenting with humidity, but I find I cannot get any humidity on account of leakage through the shingles. If I keep my shingles tight the heat will stay up there, due to the pressure exerted by the air on the outside, and that will prevent infiltration on the lower floors.

A court house was constructed with a large dome made of steel with tile. It had been painted some four or five times, but the dust would go through the paint where the steel structure came off and you could always see the iron in a year after the dome was painted. In all houses, especially warm air heated houses, a few years after the paper is put on you can see the laths through the paper. And I believe most of the heat is lost, not from infiltration through windows and walls, but due to leakage out of the roof. If you get a tight roof you will save a lot of coal and be more comfortable.

THE AUTHOR. That is a very serious situation, especially to the people who make warm air furnaces. We have had to take more than one furnace out of a church for this reason. The temperature up in the attic space above a porous ceiling would be 100 deg. and at the breathing line 40 deg.

As the last speaker has just said you can count every lath, showing how the heat that comes in is wasted through the ceiling, using it as a filter. Above the plaster ceiling an air tight T and G floor lined with building paper should be laid to prevent loss of heat through the ceiling.

No. 625

TESTS OF HUMIDITY CONDITIONS IN A RESIDENCE HEATED BY A WARM-AIR FURNACE USING RECIRCULATED AIR

By A. P. Kratz[1], Urbana, Ill.
Non-Member

THE tests reported in this paper originally formed part of an investigation of warm-air furnaces conducted by the Engineering Experiment Station at the University of Illinois under a cooperative agreement with the *National Warm-Air Heating and Ventilating Association*. This investigation of warm-air furnaces is being continued under the direction of A. C. Willard, professor of heating and ventilation and head of the department of mechanical engineering. The results from the investigation of humidity are by no means complete, but enough data have been obtained to warrant a preliminary report on the subject and this paper should be regarded as constituting such a report.

The investigation of humidity control for warm-air furnaces was started by W. E. Pratt, special investigator and research associate in the Engineering Experiment Station, in the summer of 1919. The preliminary tests were run on the furnace installed as a part of the experimental heating plant in the mechanical engineering laboratory at the University of Illinois. This plant will be designated as furnace No. 1.

The object in view in the preliminary tests was the determination of the evaporation rate of three different types of water pans for various rates of combustion, and also the effect of the location of the pan upon the evaporation rate.

The crescent shaped pan used for the preliminary tests was then removed from furnace No. 1 and installed in a furnace in a private residence, in order to study its action under service conditions and to obtain data relative to the weight of water required to produce the humidity necessary for the maximum degree of comfort. This furnace will be designated as furnace No. 2.

Owing to the withdrawal of Mr. Pratt from the research staff, it became necessary to install a crescent-shaped water pan in a furnace at another residence in order to continue the work. This furnace will, therefore, be designated as furnace No. 3. The work was continued along the same lines as before, with the additional object in view of determining

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whether a change in humidity would affect the amount of coal required to heat the house.

Description of Apparatus Used.—The plant on which the preliminary tests were run has been described in Engineering Experiment Station Bulletin No. 112, Report of Progress in Warm-Air Furnace Heating, by A. C. Willard, and is shown in the photograph in Fig. 1. It consisted of a furnace with a 27-in. firepot installed under the skeleton house

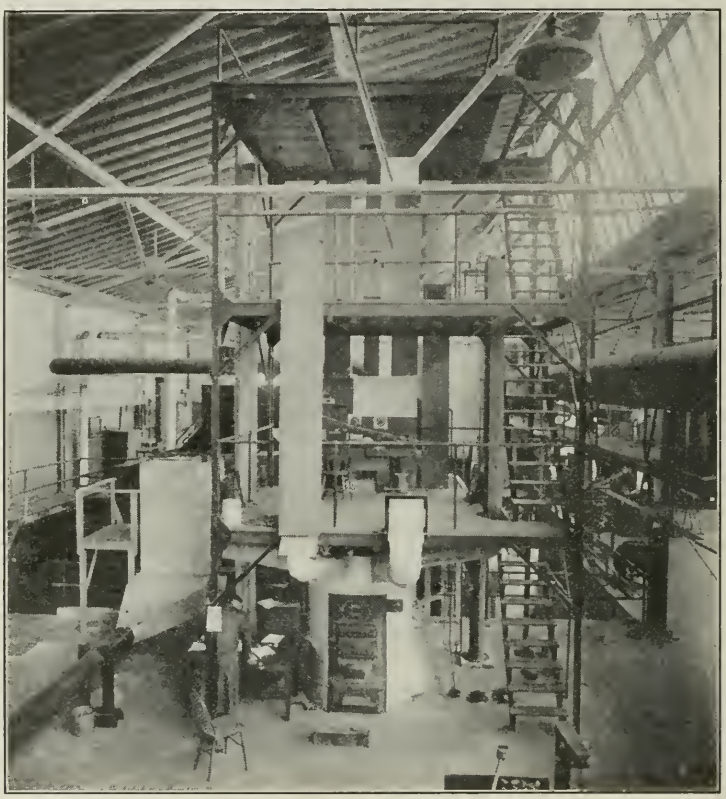

FIG. 1. VIEW OF FURNACE NO. 1 AT ENGINEERING EXPERIMENT STATION

shown. This plant had ten leaders, of which four served the first floor, four served the second floor and two served the top floor.

The first pan used was the regular cast-iron pan furnished by the makers of the furnace, and was located in the front casting just below the feed neck for the coal, as shown in Fig. 2. This pan had a total evaporating surface of 92 sq. in., 60 per cent of which extended through the front into the space between the firepot and the front casting. It is probable that all of this surface should be considered as evaporating surface although it is not as effective as though air was free to circulate over the whole surface.

The second pan, Fig. 2, was a shallow cylindrical pan placed on the

dome of the furnace. The bottom of the pan was held about 2 in. above the dome by three small legs.

The third pan, Fig. 3, was crescent shaped, with a wedge-like cross-section. It was placed just inside the casing and extended practically

FIG. 2. VIEW SHOWING FURNACE AND DOME-TYPE HUMIDIFIER

all of the way around the firepot, a gap being left for the feed neck. The object in using a pan of this shape was to intercept part of the heat radiated from the firepot, as well as to increase the effective evaporating surface over that of the other two types of pans. For the tests in the laboratory, this pan had an evaporating surface of 202 sq. in., which, how-

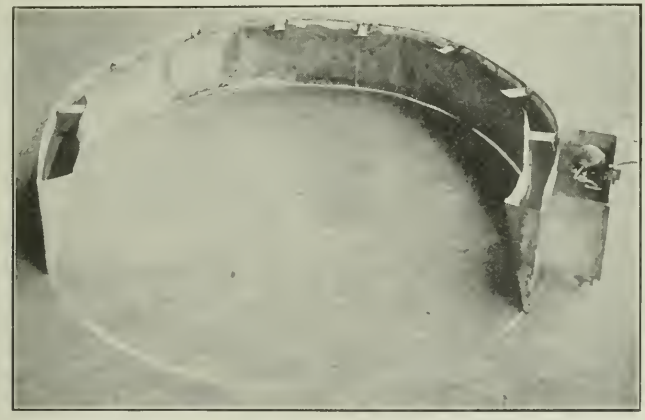

FIG. 3. CRESCENT-SHAPED WATER PAN FOR INSERTION WITHIN FURNACE CASING

TABLE 1. RESULTS OF EVAPORATIVE TESTS ON CRESCENT-SHAPED PAN IN FURNACE NO. 2—BITUMINOUS COAL

Readings taken daily from October 13, 1919 to November 9, 1919

No.	Temp. of Outside Air, °F.	Temp. at Breathing Line, °F.	Mean Temp. at Register Throat, °F.	Relative Humidity, Per Cent	Water Evap., per 24 hr., gal.	Water Evap. per sq. in. per 24 hr., lb.
1	52.0	74.6	147	50.2	1.0	0.030
2	54.0	72.3	118	57.0	1.0	0.030
3	47.0	72.9	138	47.7	3.5	0.105
4	50.0	74.0	143	45.0	1.0	0.030
5	61.0	71.3	113	50.0	2.5	0.083
6	57.0	71.5	113	57.0	1.0	0.030
7	49.0	70.7	104	55.0	0.5	0.015
8	46.0	69.7	120	54.0	3.5	0.105
9	58.0	74.0	113	58.0	0.5	0.015
10	65.0	75.0	95	65.0	0.5	0.015
11	48.5	74.0	140	55.5	0.5	0.015
12	59.0	74.0	106	65.5	1.5	0.045
13	38.0	70.3	133	51.0	2.5	0.083
14	45.0	73.5	131	50.0	3.0	0.090
15	52.0	71.5	165	53.0	2.0	0.060
16	45.0	72.0	110	48.0	1.0	0.030
17	36.0	69.0	113	44.0	1.5	0.045
18	33.0	70.0	127	44.0	3.0	0.090
19	44.0	70.0	121	50.0	1.0	0.030
20	46.0	71.0	113	50.0	1.5	0.045
21	48.0	71.0	120	49.0	1.5	0.045
22	55.0	75.0	121	48.0	1.5	0.045
Mean	49.4	72.1	122.7	52.1	...	0.049

ever, was increased to 267 sq. in. before the pan was installed in furnace No. 2. The water in the feed neck of the pan was separated from the water in the pan by a bridge, and hence the feed neck was not considered as effective evaporating surface.

For the tests on furnace No. 3, a pan similar to the crescent-shaped pan used in furnace No. 2 was constructed. In this pan, however, there was no bridge between the feed neck and main part of the pan, and the surface in the feed neck was assumed to be effective evaporating surface. The total evaporating surface was 304 sq. in. when the water level stood 1 in. from the top edge of the pan. This pan was installed in the furnace, with the straight side toward the casing. The mean distance from the inside edge to the firepot was approximately 3 in. The inner lining of the casing extended about 2 in. below the bottom of the pan. The furnace in which the pan was used had a 24-in. slotted firepot and a 22-in. grate. The diameter of the casing was 48 in. and the inside lining of the casing was 1 in. from the outside. The house had a cubical content of 22,340 cu. ft., excluding the basement and the attic, and it should be noted that the furnace was much too small for a house of this size. During the time of the tests all of the registers on the first floor, three in number, and four of the registers on the second floor were open. In all of the tests a constant water level was maintained in the pans.

Results of Evaporating Tests on Furnace No. 1.—The results of the evaporative tests run on the three types of pans in the furnace in the laboratory are shown in Fig. 4. At a register temperature of 180 deg. fahr., the evaporation per sq. in. of evaporating surface per 24 hr. was 0.98 lb. for the pan on the dome, 0.68 lb. for the crescent-shaped pan and 0.46 lb. for the regular pan. The total evaporation per 24 hr. was respectively 140, 137 and 42 lb. Thus the pan on the dome was most effective as a humidifier. Both the pan on the dome and the crescent-

TABLE 2. RESULTS OF EVAPORATIVE TESTS ON CRESCENT-SHAPED PAN IN FURNACE NO. 3—BITUMINOUS COAL

Readings taken during period from January 31, 1920 to March 12, 1920

No.	Length of Run, Hr.	Temp. of Outside Air, °F.	Mean Temp. at Register Throat, °F.	Temp. of Water in Pan, °F.	Total Water Evap., lb.	Water Evap. per 24 hr., lb.	Mean Evap. Surface, sq. in.	Water Evap. per sq. in. per 24 hr., lb.	Relative Humidity Per Cent
1	11.00	21.0	....	....	19.89	43.4	291	0.150	23.0
2	21.08	34.5	....	....	20.97	23.9	275	0.081	....
3	15.33	38.0	149	....	12.12	19.0	284	0.068	31.0
4	12.00	32.0	163	....	9.36	18.7	304	0.062	30.0
5	11.75	28.0	161	....	8.03	16.4	296	0.055	28.7
6	12.00	37.0	152	104	17.35	34.7	296	0.117	28.7
7	12.00	15.0	152	124	9.36	18.7	296	0.063	29.7
8	11.25	11.0	158	129	14.72	31.4	296	0.106	29.7
9	16.25	14.0	158	140	18.72	27.7	296	0.094	27.2
10	8.50	6.0	166	118	13.40	37.8	296	0.128	27.2
11	12.00	0.0	166	120	18.72	37.4	296	0.126	33.0
12	13.00	27.0	175	112	14.72	27.2	296	0.092	33.0
13	26.00	21.0	132	124	28.05	25.9	296	0.086	23.5
14	22.00	20.0	81	80	20.05	21.9	296	0.074	20.5
15	44.75	33.0	128	....	41.45	22.2	296	0.075	26.0
16	137.25	31.5	....	....	116.40	20.3	304	0.068	....
17	168.00	31.0	....	....	153.70	21.9	291	0.074	....
Mean								0.089	

shaped pans were much more effective than the small pan of standard type. The crescent-shaped pan, however, intercepted and used part of the radiant heat. It should be noted that all of these preliminary tests were made with the furnace fired with anthracite coal.

Comparison of Results on the Crescent-Shaped Pans.—The water evaporated per sq. in. of evaporating surface per 24 hr. for the crescent-shaped pans used in the three different types of furnaces have been plotted against register temperatures. These curves are shown in Fig. 5. Since the furnaces in the two residences were operated under service conditions and were not constantly under the control of skilled observers, the results were subject to wide variations and the relations indicated are, therefore, general. Furthermore, for all tests conducted in the laboratory on furnace No. 1 anthracite coal was used, while for the other tests bituminous coal was used. The evaporative rate for the crescent-shaped pan in furnace No. 1 rose very rapidly as the register temperatures increased. The mean value of 180 deg. fahr. register temperatures was 0.675 lb. evaporated per sq. in. per 24 hr. For the crescent-shaped pans in furnaces No. 2 and No. 3, the evaporation rate was practically constant over the whole range of register temperatures. The mean rate for furnace No. 2 was 0.049 and for furnace No. 3 was 0.089 lb. per sq. in. per 24 hr. The results for these tests are shown in Tables 1 and 2. The fact that the pan in furnace No. 3, placed midway between the casing and the firepot, shows a higher evaporation rate than the one placed against the casing, in furnace No. 2, seems to indicate that the first position mentioned is the more advantageous location for the pan.

The fact that the evaporation curve for the pan in furnace No. 1 in which anthracite coal was used rose very rapidly with the register temperature, while the curves for the pans in the furnaces where soft coal was used fell very much below this curve and did not rise with increased register temperature, led to the conclusion that the evaporation was dependent upon the radiation received from the firepot. Since hard-coal fires are much hotter than soft-coal fires, and the fuel bed is denser,

much more heat is radiated from the hard-coal fire than from the soft coal. In order to establish some relation between character of the fuel bed and evaporative rate for this pan a test was run on April 27, 1920, using hard coal in furnace No. 3. The results of this test are given in Table 3. The mean evaporation rate at a register temperature of 193.5 deg. fahr. was 0.292 lb. of water per sq. in. per 24 hr. At the same register temperature, when soft coal was used, the evaporation was 0.089 lb. per sq. in. per 24 hr. Thus the evaporation for anthracite was 3.4 times that for bituminous coal. This is shown in the curves in Fig. 5 in which the point for hard coal fell above the curve shown for furnace No. 3.

Analysis of the results in Table 3 prove that the evaporative rate changed with a change in the intensity of radiation. The radiation

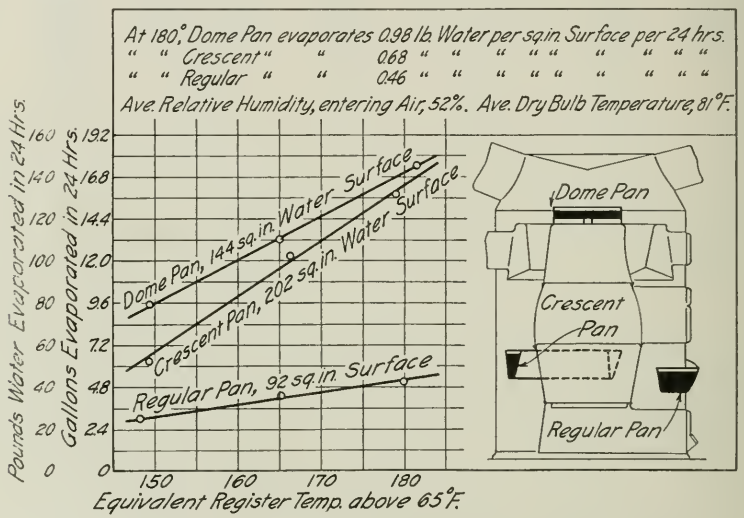

FIG. 4. RESULTS OF EVAPORATIVE TESTS ON WATER PANS

reached its maximum intensity during the hour between 11:00 and 12:00 a. m., when the evaporation was 0.407. At this time the total mass in the fuel bed had become incandescent and had not lost much weight. Also very little ash had been formed. From this time on, until 10:00 p. m. the rise in the temperature of the air remained about constant, or varied on either side of a mean, while the evaporation gradually decreased. This was due to the fact that the weight of fuel was constantly becoming less while the temperature of the fuel bed remained about the same. At the same time, ash was being formed on the sides of the firepot, thus reducing the radiant effect of the incandescent fuel. After firing a fresh charge at 9:30 p. m., the evaporation decreased. As this black bed became incandescent, the evaporation increased until 1:00 a. m. when it reached a value nearly the same as the one toward the close of the first firing period. This indicated that the ash had an appreciable effect in reducing the radiation from the fuel bed. The first three hours and the readings subsequent to the second firing have not been included

in the average evaporation for the anthracite, in order to obtain values comparable with the fuel bed in normal running condition. This mean value is higher than the mean of all readings, while the mean air temperature remains nearly constant.

Effect of Pan on Increase in Relative Humidity.—The results of the tests run before the installation of the crescent-shaped pan in furnace No. 3 are given in Table 4 and the results of the tests run after the installation of the pan are given in Table 5. The immediate effect of the pan was to increase the mean relative humidity from 23.6 per cent to 28.4 per cent, or an increase of 20.3 per cent over the original relative humidity. The mean temperature difference between air at the breathing line and outside air was 44.9 deg. fahr. before installing the pan, and 46.0 deg. fahr. afterward. The register temperatures, however, showed

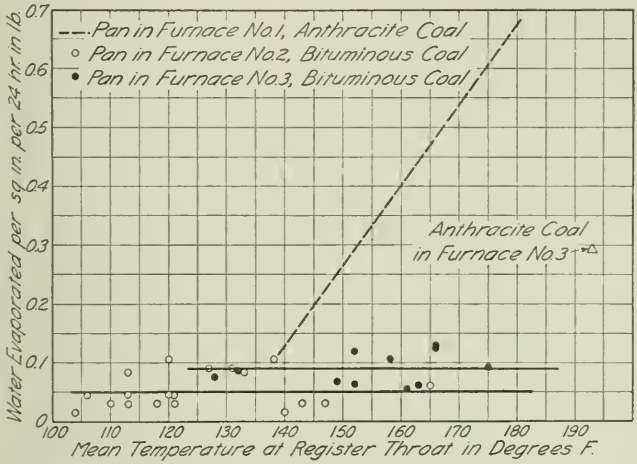

FIG. 5. COMPARISON OF RESULTS USING CRESCENT-SHAPED PAN

a marked decrease after the installation of the pan. The mean register temperatures before the installation were 177.0 deg. fahr. and 147.5 deg. fahr. afterward. That is it required register temperatures averaging 29.5 deg. fahr. less in order to give a range in temperature 1 deg. fahr. greater. This is attributed to the higher specific heat of the water vapor in the air, which enabled the weight of air circulated to carry the same amount of heat at a lower mean temperature. This is significant in view of the fact that from the standpoint of comfort it is desirable to introduce the air for heating at as low a temperature as is consistent with the maintenance of proper temperature at the breathing line.

Effect of Increased Humidity on Coal Consumption.—The coal consumption per 24 hr. for the series of tests run before and after installation of the crescent-shaped water pan in furnace No. 3 is shown in Fig. 6. The coal consumption is plotted against the temperature of the outside air. The mean temperature at the breathing line was approximately the same for both series. For the series without the pan this temperature was 67.7 deg. fahr. and for the series with the pan it was 70.8 deg. fahr.

TABLE 3. RESULTS OF EVAPORATIVE TESTS ON CRESCENT-SHAPED PAN IN FURNACE NO. 3—SCRANTON ANTHRACITE COAL

No.	Time	Temp. of Outside Air, °F.	Mean Temp. at Register Throat, °F.	Mean Temp. at Cold Air Face, °F.	Mean Temp. Rise, °F.	Temp. of Water in Pan, °F.	Water Evap. per sq. in. per 24 hr.	Relative Humidity Per Cent
1	8:45 a.m.	....	....	....	....	....	....	....
2	9:00	48.0	154.0	63.0	91.0	....	....	....
3	10:00	48.0	207.0	68.0	139.0	162	....	69.0
4	11:00	49.0	167.0	70.0	97.0	167	0.224	44.0
5	12:00	47.0	215.3	71.5	143.8	178	0.407	53.0
6	1:00 p.m.	47.0	181.5	74.0	107.5	161	0.328	43.0
7	2:00	48.0	186.0	73.5	112.5	165	0.336	40.0
8	3:00	46.0	197.0	74.5	122.5	165	0.312	43.0
9	4:00	45.0	185.0	74.0	111.0	160	0.328	45.0
10	5:00	44.0	199.5	75.5	124.0	163	0.280	48.0
11	6:00	43.0	225.0	76.5	149.0	165	0.312	40.0
12	7:00	44.0	186.4	65.0	121.4	148	0.264	34.0
13	8:00	43.0	171.0	74.0	97.0	144	0.208	37.0
14	9:00	44.0	185.0	74.0	111.0	138	0.200	34.0
15	10:00	44.0	197.0	74.5	122.9	128	0.240	35.0
16	11:00	....	191.0	75.0	116.0	144	0.208	....
17	12:00	....	169.0	74.5	94.5	146	0.136	....
18	1:00 a.m.	....	194.0	74.5	119.9	146	0.208	....
Mean of Readings 5 to 15		45.0	193.5	73.5	120.0	156.0	0.292	41.0

NOTE.—Mean evaporating surface = 300 sq. in.
Fire was not incandescent during starting period from 8:30 to 11:00 a. m.
Fresh coal fired at 9:30 p. m. Fire black then.

TABLE 4. HUMIDITY RESULTS, BEFORE INSTALLING CRESCENT-SHAPED PAN IN FURNACE NO. 3—BITUMINOUS COAL

Readings taken daily from December 2, 1919 to December 25, 1919

No.	Temp. of Outside Air, °F.	Temp. at Breathing Line, °F.	Temp. Diff. Inside and Outside, °F.	Mean Temp. at Register Throat, °F.	Mean Basement Temp., °F.	Temp. at Throat of Recirculating Duct, °F.	Relative Humidity, Per Cent	Coal Fired per 24 hr., lb.
1	18.5	65.5	47.0	239	....	....	20.5	....
2	13.0	64.9	51.9	166	....	60.0	20.0	....
3	30.0	70.0	40.0	131	....	63.0	25.0	....
4	30.0	72.5	42.5	156	....	62.0	25.0	....
5	33.5	74.0	40.5	193	....	64.0	27.3	....
6	28.0	68.5	40.5	166	....	63.0	25.0	....
7	25.0	69.1	34.1	136	....	60.0	24.8	200
8	14.0	66.8	52.8	152	....	57.0	21.5	213
9	3.0	56.0	59.0	215	....	46.0	21.5	225
10	28.0	68.1	40.1	176	....	54.0	20.0	200
11	36.5	75.3	38.8	154	....	62.5	23.0	144
12	11.0	65.8	54.8	165	....	52.0	20.5	206
13	20.0	64.5	44.5	238	....	49.0	....	219
14	13.0	62.5	49.5	149	51.0	48.5	25.0	162
15	20.0	67.5	47.5	112	54.0	54.0	21.0	181
16	26.0	67.5	41.5	120	54.0	57.0	28.0	150
17	32.0	74.0	42.0	185	59.0	61.0	27.0	131
18	26.0	66.0	40.0	121	....	....	....	138
19	32.0	68.5	36.5	149	55.0	57.0	26.0	125
Mean	22.8	67.7	44.9	164.5	54.7	57.0	23.6	177

NOTE.—Maximum register temperature observed—294 deg. fahr.
Minimum register temperature observed—84 deg. fahr.

The points do not fall on a well marked curve and the mean tendency only can be represented by a broad band drawn as shown in Fig. 6. There seemed to be no marked change in the coal consumption after the installation of the water pan. A single curve has, therefore, been used to represent all of the points.

CONCLUSIONS

1. More water is evaporated per sq. in. of surface from a pan supported on the dome of a furnace than from pans placed around, but not touching, the firepot.

2. If a crescent-shaped pan is used surrounding the firepot the best location for this pan is midway between the casing and the firepot.

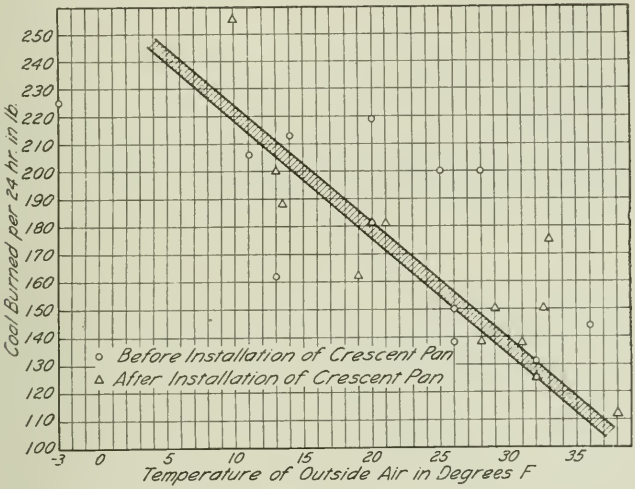

FIG. 6. EFFECT OF HUMIDITY ON COAL CONSUMPTION

3. The crescent-shaped pan is not very effective in increasing the relative humidity. In the tests with the soft-coal fuel bed, the actual change was only from 23.6 per cent to 28.4 per cent. At no time was frost or condensation observed on the outside glass with either of the above humidities.

4. Increasing the relative humidity makes it possible to maintain the same difference in temperature between the air at the breathing line and the outside air with the use of lower register temperature.

5. The evaporation per sq. in. per 24 hr. from the surface of the water in a crescent-shaped water pan, surrounding but not touching the firepot, varies with the intensity of the radiation from the fuel bed. This in turn is a function of the temperature and density of the fuel bed. The heat radiated from a hard-coal fuel bed is much greater than that radiated from a soft-coal fuel bed.

6. The coal consumption for a furnace in service is not materially decreased when the humidity is increased and the temperatures at the breathing line maintained constant.

TABLE 5. HUMIDITY RESULTS AFTER INSTALLING CRESCENT-SHAPED PAN IN FURNACE NO. 3—BITUMINOUS COAL

Readings taken during period from January 31, 1920 to February 28, 1920

No.	Temp. of Outside Air, °F.	Temp. at Breathing Line, °F.	Temp. Diff. Inside and Outside, °F.	Mean Temp. at Register Throat, °F.	Mean Basement Temp., °F.	Temp. at Throat of Recirculating Duct, °F.	Coal Fired per 24 hr., lb.	Relative Humidity, Per Cent	Water Evap. per lb. of coal, lb.
1	21.0	76.0	55.0	....	....	....	23.0	162	0.268
2	38.0	73.3	35.3	149	....	62.0	31.0	112	0.170
3	32.0	76.3	44.3	163	....	65.0	30.0	125	0.150
4	30.0	70.1	40.1	151	....	61.0	32.5	150	....
5	28.0	73.5	45.5	161	....	61.5	30.7	138	....
6	32.5	75.5	43.0	152	62.0	64.0	28.7	150	0.170
7	13.0	69.8	56.8	158	....	54.5	29.7	200	0.125
8	10.0	66.7	56.7	166	52.0	48.0	27.2	256	0.128
9	13.5	64.1	50.6	175	....	50.0	33.0	188	0.172
10	31.0	70.3	39.3	155	....	59.0	33.5	138	....
11	21.0	70.0	49.0	132	....	58.0	23.5	181	0.143
12	20.0	61.8	41.8	81	....	52.0	20.5	181	0.121
13	33.0	75.3	42.3	128	....	64.5	26.0	175	0.127
Mean	24.8	70.8	46.0	147.5	57.0	58.2	28.4	166	0.157

NOTE.—Maximum register temperature observed—221 deg. fahr.
Minimum register temperature observed—77 deg. fahr.

7. The air conditions were decidedly more satisfactory to the occupants of the house after the installation of the crescent-shaped pan than before, notwithstanding the fact that the actual increase in relative humidity was small.

DISCUSSION

W. J. BALDWIN: Apropos of a question that often comes up relative to the cost of humidity in houses in winter, when the water has to be evaporated at the expense of the fuel, I desire to give my reasons for the position I have taken, when I said "the extra cost of the fuel was great enough to be taken into consideration when the question of humidifying the air of a residence in winter was considered."

I do not desire, however, to convey the idea that the extra cost is prohibitory; it may mean money well laid out, but the real thing to get into the mind of the ordinary steamfitter, is the fact that *artificial humidity* costs money. By artificial humidity I mean, either the evaporation of water in open pans or any other way. Many years ago, in my first book on Steam Heating (1878), I made the statement "that to evaporate a pound of water from a pool in the street, required just as much heat, as to evaporate a pound of water in a boiler," and this applies to the evaporating-pan in the furnace, or to the steam jet from the boiler.

A twelve-room house was mentioned as an example of an ordinary city residence, and a ton of coal, per room, per winter, in the neighborhood of New York, has been considered about right for the fuel required. This means an average New York winter (or cold season) of from 182 days to 200 days, and is approximately correct with a heating apparatus without humidifying, be it steam or hot water or a furnace.

Therefore, it is reasonable to assume that a twelve-room brick house in a New York City block (not a corner house) requires an average of 12 tons of anthracite coal for a winter of 200 days (no humidifying), 12 tons of dry coal weighs 24,000 lb. and this calls for 120 lb. of coal per

day, or 5 lb. of coal per hour, through day and night, representing 50,000 B.t.u. available hourly.

If each of the rooms of the house averages 1,000 cu. ft. of air space, and we change the air only twice an hour, then we have to humidify 24,000 cu. ft. of air per hour for the house if we desire to keep it at some *reasonable* standard of humidity, say 50 to 60 per cent of saturation. Air at 70 deg. fahr., when saturated, holds very nearly 8 grains of moisture per cubic foot, and 50 per cent of saturation, will be *over* 4 grains, while the air outside in winter weather, contains only between 1 and 2 grains of moisture. Thus the air may be considered practically dry, when admitted to the house (say 1 grain to the cubic foot). Therefore, it is not unreasonable to assume that we must add 4 grains and over of moisture, to each cubic foot of air entering the house in an hour, if we are going to keep the air in the house at anything near 50 per cent of saturation.

This means that we have to evaporate 96,000 grains of water in an hour, or very nearly 2 gal. of water per hour, for the house and conditions assumed; or more accurately 14 lb. of water, the equivalent of about 14,000 B.t.u. Therefore the quantity of fuel for heat alone per hour is 50,000 B.t.u. and for humidifying, 14,000 B.t.u. per hour, which is considerably over 25 per cent of the amount of fuel required for heating alone (not considering the warming of the vapor of water to the house temperature).

At the meeting of New York Chapter of the Society on December 19, I inadvertently conveyed the idea that the *ratio* of the cost of humidifying was greater than the above. I had in mind experiments made in my own house, where I found the ordinary winter conditions within the house, furniture, carpets, walls, etc., seized on the moisture added to the air, so rapidly, that I had to admit about twice the quantity of moisture before outlined, to keep a condition of humidity of about 50 per cent.

Perhaps when *a standard of humidity* is kept constantly and systematically, the cost can be kept as low as 25 per cent of the cost of warming alone. But for experimental runs of even as much as 24 hours' duration, the dry woodwork, the dry plaster, etc., all seized on the moisture so rapidly, that a rough estimate of even 50 per cent additional fuel is not improbable.

My reason for presenting this matter in this way, is to let it be understood that *humidity costs some money* when applied to the humidification of our homes. To some, the 25 per cent additional cost for fuel may be a bagatelle, if they consider they are improving their health, though I am not convinced that a damp atmosphere is so necessary to health, in North America, as many of us think.

Dr. John S. Billings, Sr., U. S. A. (an authority in ventilation) when in Yuma, Arizona, told me that he had to keep his lead pencils standing in a glass of water when he was writing; he also said ink was out of the question, as the ink would dry so rapidly in a pen, he could not use it, and if I remember rightly he said that the health of the troops stationed there (in almost the driest atmosphere in the world) was excellent.

The Chief Medical Officer of Bermuda, about 40 years ago, told the writer that the humid climate of Bermuda did not help consumptives, but he also said, it was good for those who had only bronchitis. His opinions were based on statistics of the British Army.

DR. WOLFF FREUDENTHAL: We need more humidity in our homes. We need humidity firstly for our own well-being and then for the preservation of our furniture, our books, our plants, etc. The minimum requirement should be 40 per cent relative humidity. We have heard today that relative humidity was raised from 23 to 28 per cent. This is not enough—40 per cent ought to be the minimum amount. Now is just the time to test the validity of our humidifying apparatus, while the outside temperature is low.

I had the pleasure of speaking before this Society when it last met in New York and I mentioned the fact that my attention was drawn to the lack of humidity in our houses more than 20 years ago. We need more humidity to be healthy. These days you see many people walking around with bronchitis and all sorts of affections in the throat and nose. They feel dull. As soon as they take a hot-water plunge, inhale warm water, they feel entirely different. You feel the same depression after you come out of the opera house, which is comparatively better ventilated, so far as I know, than other institutions of that class. The reason is that the atmosphere there is too dry. Bermuda may not help one class of consumptives, but may help another. More than 100 years ago the great French physician Laennec sent his consumptive patients to the Mediterranean, directly on the seashore, and had success in many instances. Fifty years later a German physician, Brehmer, sent his patients to the mountains and a great many cases improved, too. One class of these patients needs a dry climate with a high altitude, while the other requires a humid climate.

At any rate, no matter what disease any patient is suffering from or how healthy he may be, we all need more humidity in our *homes*. One gentleman, I believe from Toronto, figured that in a twelve-room house we should not have a quart of water but 50 or 60 gallons of water every day to evaporate; that is for a moderate house of twelve rooms.

I think our system of heating is entirely wrong. I cannot change it, gentlemen, but we all are waiting for you, because you are the pioneers in that work and you understand better than we physicians do. At any rate, we are looking forward and welcome every progress that is made in humidifying the atmosphere in our homes.

F. R. STILL: I think that the hypothesis upon which is founded the argument as presented by Dr. Freudenthal is correct except in this respect. It is a well-known fact that in a house where no attempt has been made to humidify the air, instead of carrying the temperature at about 70 deg. fahr. it is more generally around 80 or 84 deg. If the relative humidity should be raised to about 50 per cent at this latter temperature, we could not stand it. With a relative humidity of 45 to 50 per cent, we would all be perfectly comfortable with a temperature of 64 to 66 deg., which will save just about as much coal as the lower temperature would amount to, less the cost to heat the water required to produce the desired

humidity. In other words, approximately all the heat we put into the water goes into the house.

We have made an effort many times to determine the cost of operation on many types of buildings, but it is almost impossible because we never find two buildings alike. In almost any city there may be found two schools, exactly alike, one being equipped with a humidifying apparatus and the other without. The one that is humidified may only cost about 75 per cent of the fuel which the other costs. The 25 per cent difference, however, may be due to the man who operates the plant or any one of many other things. But it does look, as near as we can find out, that with a 56 deg. wet bulb temperature, the dry bulb must be so much lower that there is a very material saving of fuel and much greater comfort obtained by the occupants.

A common mistake made when discussing this subject is talking about relative humidity. We should all try to avoid it. What we want to talk about is absolute humidity, or what is the wet bulb temperature. We are all fairly well convinced that 54 to 56 deg. wet bulb temperature is the comfort and health temperature.

It is now, I believe, rather a question among the medical fraternity whether it is the relative humidity, or whether it is sunshine, that has the greater effect in the restoration of tubercular patients to health, assuming that the air is equally pure when making comparisons.

We do know that clean air has a marked effect. Removing dust and dirt from the air, as was explained by Dr. Hill yesterday, reduces the bacterial count just in proportion to the amount of dirt removed; hence by increasing the humidity, it must follow that there is a lessening of the tendency for bacteria and germs to be carried from one person to another.

The great drawback to adequately humidify a residence is that there are so few effective devices which are automatic in operation; consequently they fail to give satisfactory results because they are not kept in continuous operation, neither are they kept clean. Cleaning a humidifying apparatus is quite as necessary as it is to clean an air washer. It is very difficult to clean most of the devices I have seen; in fact, this feature seems to have been ignored in most of them. Furthermore, I have never seen but one simple type of device which looked to me as though it had any real merit. It consisted of a pan located above the crown of the furnace, somewhat along the lines recommended by Mr. Kratz in his paper. Then there was a pipe leading from this pan to another and deeper pan outside the furnace jacket, with a ball float in it, operating a valve in a water supply pipe. As the water evaporated from the inside pan, it lowered the level in the outside pan; the float then opened the valve to let in more water, thus keeping the level constant. The principal objection to this device is that when one of the pans is put in above the crown of the furnace, it can never be gotten at, hence it will never be cleaned. It is sure to become very dirty in a little while.

I think it is a mistake to even suggest that it adds to the operating cost to humidify air, considering the beneficial effects resulting from it. People are so prone to avoid the use of everything which costs them an

indefinite amount to operate, that they are likely to draw the wrong conclusion from this discussion if it is not at once convincingly established as it can be that it costs less, instead of more to operate a heating plant which is properly equipped with an approved humidifying apparatus.

When our bodies are surrounded by a dry atmosphere, more heat is taken from us, due to the latent heat of evaporation from the skin. Hence we feel chilly at temperatures in the winter which are perfectly delightful in the summer time when there is more moisture in the air and consequently less evaporation and less heat taken from the skin. If we will maintain 56 deg. wet-bulb temperature in our homes, we will be far more comfortable at dry-bulb temperatures of 64 to 66 deg. than we are at 84 deg. dry-bulb temperature when the air which surrounds us is usually less than 10 per cent saturated.

A MEMBER: In 1914 Mr. Flobun, who was an engineer for Morris & Company in Chicago, resigned on account of bad health and went to Florida, and while he was in Florida he conceived the idea that it was the atmospheric conditions in Chicago that had affected his health. He accordingly worked on a humidifier to be used in connection with a radiator, and I see in an advertisement in this January issue of the JOURNAL that it bears his name. He used spun glass and after many experiments he found out that spun glass was the most sanitary substance he could use. He applied his apparatus to a radiator and anyone that has a house fitted with radiators will find it a very cheap installation as without special tools he can put it up and will find it is very satisfactory.

P. J. DOUGHERTY: In my opinion it is not advisable to place the humidifier on top of a warm air furnace radiator, although it will evaporate more water at that point. In case the humidifier should leak or should boil over, or overflow, it is liable to crack the hot furnace castings. It also may interfere with the air passing through the center openings of the radiator. The most practical place is to locate it opposite the firepot where there is usually ample space to place as large a one or as many small ones as needs to evaporate the water required. This location also intercepts considerable radiant heat that might otherwise be lost by passing out through the casings. The amount of water required to be evaporated is in practice much lower than that theoretically calculated, since most warm air installations recirculate the air within the house. The normal infiltration around windows and doors usually supplies far more fresh air than is required.

A MEMBER: I occupy a house about 22 by 35 ft. of about nine rooms which is heated by hot air, taking the air from outside. My experience may be interesting—even though some others disagree—that in weather like this, with a temperature averaging below 10 deg. we evaporate from our humidifying pan about 9 gal. of water in 24 hours. From this maximum we run down to a minimum in very mild weather, early fall and late spring, of perhaps a gallon, especially if the air is moist outside. After a few trials the people in the house very soon realize when the pan is nearly dry by the quality of the air they are breathing. It seems to me that it is a question not alone of the health of the occupants of the house, but also of the wet bulb temperature that is such an important

factor in the feeling of the air. Some people are not very sensitive to it, but I find that those who have lived in air heated houses almost invariably are.

A. A. ADLER: If one goes outdoors and breathes the air he adds both heat and humidity. I cannot understand why when air is brought into the house water must be added to it before it is inhaled.

W. G. R. BRAEMER: Dr. Adler refers, I think, to the relative humidity of outside air. The relative humidity during cold weather is usually very high yet, the absolute humidity or actual moisture content of the air is very low. The amount of moisture that air will hold depends upon the temperature. For instance, air at zero and saturated will hold about 0.5 gr. per cu. ft. Air at 70 deg. fahr. will hold about 8 gr. per cu. ft. at saturation. Then if air is taken from outside at, say, zero degree and completely saturated, and it is desired to maintain 70 deg. and 50 per cent relative humidity in the room, 3.5 gr. of moisture will have to be added to each cu. ft. of air handled.

A. A. ADLER: That does not explain it at all. We breathe in the air and exhale it. Now in that process we are adding a certain amount of moisture to that air and adding a certain amount of heat to it. If we take air from indoors the body has simply been relieved of adding heat. The cycle is somewhat as follows: Cold air is drawn in and heated by the nose, throat and lung passages, when the relative humidity is lowered due to the high temperature, consequently moisture is absorbed. On exhaling, however, the air is heated to body temperature or thereabouts and has a high relative humidity. On passing over the same surfaces it traversed previously, it returns heat and humidity. Therefore while coming in it has a drying and cooling effect, but while going out, the conditions are reversed.

O. W. ARMSPACH: The amount of moisture taken from the air passages, that is, the moisture removed from the body in the expired air is a function of the absolute humidity of the air. There is as much moisture removed from the body in the expired air outdoors, where the air may be 10 deg. and 90 per cent humidity, as there would be in a room where the temperature was say 70 degrees and only 8 per cent humidity. It may be demonstrated that the amount of moisture removed is practically the same in either case. The moisture removal cannot be expressed as a function of relative humidity unless humidity is given with some dry bulb temperature and relative humidity is not a direct measure of the moisture drawn from the mucous membrane.

Although the moisture removal in the two conditions given above is equal no doubt the irritation on the mucous membrane is not the same. There may be a difference as to the point in the membrane at which moisture is removed. If at low humidities all moisture is taken from the nose you are going to feel an irritation. On the other hand it may be that at low humidities moisture is taken from the nose and also further back into the air passage, thus accounting for a greater irritation under low relative humidity conditions.

F. R. STILL: I think the answer to Mr. Adler's question is this: What makes him feel good outside is that one seldom finds the relative

humidity below 50 per cent. Air fully saturated at 32 deg. carries 2 gr. of moisture. Air heated up to 200 deg. will carry 202 gr. Now if we bring in air from the outside and heat it up to 200 deg., we will have only one per cent relative humidity. If we want to bring the heated air up to 50 per cent relative humidity we must add 100 gr. of moisture to it. In the room we now occupy there is probably no difference in the absolute humidity from the moisture outside, excepting for the small amount of moisture added by evaporation from our bodies; hence—there is practically no difference in the absolute humidity in here and outside, but the temperature being perhaps 70 deg. higher than it is outdoors, the relative humidity is perhaps only 10 per cent inside, which has an irritating effect that does not prevail outside when the relative humidity is 50 per cent; any apparent irritation is at once eliminated inside, when the dry bulb is maintained at about 66 deg. and the wet bulb temperature is 56 deg., which is about 50 per cent relative humidity.

A MEMBER: A man breathing the air outdoors is warming up that air as he inhales. That might produce the reaction which would give him that good feeling which he would not get in breathing in air at 60 or 72 deg.

A. A. ADLER: The air that affects a man's throat is not exhaling; it is inhaling, and the air outside at high relative humidity does not dry his throat.

A MEMBER: When he breathes in the outside air he does not get that drying effect. When he comes on the inside the difference is greater and he has the drying effect as he breathes in.

DR. WOLFF FREUDENTHAL: The question raised by Dr. Adler seems so simple to me that I believe almost everybody ought to understand it. When we are outside on a cold winter day, we breathe that air, that is, we saturate the mucous membranes of the nose, throat, and bronchi with the moisture existing outside. Now when we enter a room, conditions are entirely changed. The air does not contain 60, 80, or more per cent of moisture, but somewhere around twenty. With a temperature of from 70 to 80 deg. fahr. this is very little and too dry for normal breathing purposes. The consequence is that instead of the necessary and much needed humidity being inhaled with every breath, so much is given off to the surrounding atmosphere. A drying process sets in that will affect the membranes of the nose and throat sooner or later. Unless fresh humidity is brought into the room in some way, the physiological limit will be reached soon and the person begins to ail. This we should prevent, if we can.

No. 626

THE SEMI-ANNUAL MEETING

1922

American Society of Heating and Ventilating Engineers

THE unusual practice of holding the Semi-Annual Meeting of the Society in two cities proved to be a happy one attracting as it did the largest attendance of any meeting to date. The first two days, June 6 and 7, were spent in Buffalo where the Society was entertained by Western New York Chapter. From there the visitors took the boat for Detroit and continued the sessions at the Hotel Wolverine as the guests of Michigan Chapter.

The three professional sessions in each city were taken up with technical papers of exceptional merit and the well-attended sessions indicated the interest of all who were present. The subjects covered a wide range of work in heating and ventilating; the animated discussions following each paper showed that the members were keenly appreciative of the work of the authors. The Research Laboratory contributed six technical papers demonstrating that a great amount of experimental work is being accomplished for the Society. The many problems before the Laboratory when solved will give results that will greatly benefit the world at large.

Reports from several committees such as the Committee on Code for Installation of Warm Air Furnaces, Standard Code for Testing Fans, and Steam and Return Main Sizes, created much discussion, and all of these Committees will be enabled to continue their work and report much interesting data by the next Annual Meeting.

Any doubt as to the advisability of dividing the Meeting between two cities was dispelled by the large attendance in both places. The membership was well represented, the largest delegations being from New York, Michigan, Illinois, Pennsylvania and Ohio. Minnesota sent one member, Colorado sent two, Texas, one, and Canada, 15.

The tremendous amount of registration work necessary to take care of 267 members and guests in Buffalo, and the 233 in Detroit, was handled acceptably in both places and in the matter of entertainment the local committee in both cities vied with each other in filling up the spare time of the members, as well as keeping the women delightfully entertained at all times.

PROGRAM OF THE SEMI-ANNUAL MEETING, 1922.

FIRST SESSION—BUFFALO
Tuesday, June 6, 10 a. m.

BUSINESS SESSION:
- Address of Welcome
- Response by President Jay R. McColl
- Annual Reports of Chapters
- Report of Committee on Code for Testing Fans
- Report of Committee on Installation of Furnaces
- Report of Committee on Guide Publication
- Paper:
  - Relation of Architect and Engineer, by E. T. Bradley

SECOND SESSION—BUFFALO
Tuesday, June 6, 2 p. m.

PROFESSIONAL SESSION:
- Papers:
  - Modern Fan Design, by H. F. Hagen
  - Comparative Tests of Automatic Ventilators, by J. P. Calderwood, A. J. Mack, C. J. Bradley
  - Water Pipe Sizes for Plumbing Fixtures, by W. S. Timmis
  - Corrosion in Water Carrying Vessels, by J. E. McDermet

THIRD SESSION—BUFFALO
Wednesday, June 7, 10 a. m.

RESEARCH SESSION:
- Report of Committee on Research
- Papers:
  - Some Research Conclusions, by F. Paul Anderson
  - Body Temperatures and Their Measurement, by Dr. W. J. McConnell and F. C. Houghten
  - Standard Method of Testing Dust Removal Efficiencies of Air Washers, by O. W. Armspach and Margaret Ingels
  - With the Development of a New Method of Making Air Dust Determinations, by F. Paul Anderson and O. W. Armspach
  - Dustiness of Air in Granite Cutting Plant, by S. H. Katz, L. J. Trostel

FOURTH SESSION—DETROIT
Thursday, June 8, 10 a. m.

SCHOOL VENTILATION SESSION:
- Papers:
  - Ventilation Fundamentals—Where Does the Society Stand? by Dr. E. V. Hill
  - The Intermediate Schools in Detroit, by H. W. Anderson

FIFTH SESSION—DETROIT
Friday, June 9, 10 a. m.

PIPE SIZE SESSION:
- Report of Committee on Steam and Return Main Sizes
- Pipe Size Data from the Guide
- Papers:
  - Capacity of Steam Heating Mains as Affected by Critical Velocities of Condensate Mixtures, by F. C. Houghten and L. Ebin
  - Critical Velocity in One-Pipe Steam System, by F. E. Giesecke
  - Theory for Flow of Condensation in Return Mains, by R. V. Frost
  - Temperature Control by Fractional Distribution of Steam, by J. A. Donnelly

SIXTH SESSION—DETROIT
Saturday, June 10, 10 a. m.

PROFESSIONAL SESSION:
- Papers:
  - Recent Developments in Furnace Heating, by F. R. Still
  - Derivation of the True Thermal Conductivity Coefficient from Overall Test Results, by P. Nicholls
  - Eugene Peclet as a Heating and Ventilating Engineer, by P. Nicholls
  - Air Conditioning for Sausage Plants, by M. G. Harbula

No. 627

FAN BLOWER DESIGN

By H. F. Hagen, Boston, Mass.
Non-Member

FAN blowers have usually been regarded as inefficient machines. This impression has become somewhat general for two reasons. The first is that a commercial fan is not a difficult apparatus to build and consequently the fan business has attracted a number of small concerns who entered it with very little consideration for engineering development. They were able to build fans usually for special purposes that delivered air and seldom were any of these fans under 50 per cent efficiency. Almost any centrifugal fan in a spiral casing will have an efficiency better than 50 per cent. The second reason is that often the size of the fan is governed by cost instead of by the work required of it so that more often than not the fans will work at a point very much below their maximum efficiency.

This general opinion has never been justified. The early pioneers in the fan blower industry built fans that developed a 70 per cent efficiency. They were paddle wheel type with few blades. They were designed empirically it is true, but there must have been a great deal of careful and scientific experimentation to raise the efficiency from 50 to 70 per cent with such an apparently crude design by only a refinement of proportions. These early fans enjoy a large sale today and for many purposes have not and probably cannot be improved upon.

The performance of these paddle wheel fans has been somewhat discouraging to designers. Many designs with elaborate blade shapes, shock free entrance, diffusion blades and other refinements failed to equal the efficiency of the old straight bladed paddle. The designers work was not entirely in vain however as types were developed with higher speeds, smaller size, and greater pressures which achieved commercial success. But there was no improvement in efficiency, an attainment very dear to the heart of the designer, and perhaps the one measure of real progress.

The results of such work led to a belief which found its way into authoritative print that about 70 per cent was the maximum efficiency that could be secured. Fan engineers seemingly accepted the conclusion that fans could not be designed other than experimentally. It is to be regretted that such considerations were allowed to hinder and discourage the application of mathematical theory to fan design. In no kind of power conversion machines is the absence of such theoretical guidance more

[1] Research Engineer, B. F. Sturtevant Co.

Paper presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

clearly demonstrated than in the history of the fan industry. Fans have been constructed in any number from designs based not on the laws of fluid flow and fan action but on the curves formulated in the elementary text-books on analytical geometry.

An excellent example of this kind of work comes to mind. A new fan was put on the market. Its advertising literature laid great stress on the wonderful blade shape employed which was a combination of an Archimedean spiral and a helix. It was explained that an Archimedean spiral moved out equal radial distances and a helix moved forward equal axial distances for uniform angular rotation. What could be nicer than to combine these two interesting curves in a fan blade. The idea of course was to move the air in axially and out radially with a uniform velocity. The fact that the fan impeller could do no work without giving the air a rotation and that with such rotation any virtue of uniformity based on the spiral and the helix would be entirely lost was quite overlooked. The efficiency of the fan was around the usual 55 per cent.

There have been all kinds of these geometric fans designed and built both in this country and abroad. Among them have been hyperbolic fans, involute blades, conoids, parabolic inlets and many more. Sometimes with reason, often without and never with sound science. It is hoped that some of these names were selected for their advertising purposes and were not considered fundamental in the fan design. However even if such was the case the trend in the industry is clearly shown and indicates an absence of earnest search for real guidance.

It was the writer's good fortune when he first entered the fan industry to have been closely associated with George H. Gibson. Mr. Gibson had long been interested in hydraulics and aerodynamics and had collected extensive data on these subjects. This information he very kindly placed at the disposal of the writer. Mr. Gibson further suggested that the solutions of the flow of the ideal fluid of classical hydrodynamics might form a rational guide to the design of fans.

The writer has designed several fans using as a basis and guide the theories of hydrodynamics. The results have been so uniformly gratifying that this paper is presented describing the methods by which the forms and proportions of the latest design were determined. The exceptional performance of the fan prompted the writing of this paper and must be the final argument in favor of the methods employed.

The fluid dealt with in the mathematical theory of hydrodynamics is an ideal one being incompressible, inviscous and flowing between frictionless boundaries. It is probably due to the ideal nature of the discussions involving this fluid that wider application of the theories governing its flow has not been made. However there have been so many indications showing the similarity in form of flow of air and of the ideal fluid that it would seem folly to ignore the wonderful solutions of fluid flow problems thus afforded.

In connection with the theoretical discussion of the airplane the Helmholtz-Kirchoff solution of the flow past a plane barrier with the discontinuous streamline was given considerable prominence. By these means the pressure produced on such a barrier was brought into agreement with experiment and the error of the Newtonian solution demon-

strated. That the actual flow of the air takes place substantially as outlined is readily demonstrable by smoke tests.

Recently a solution has been secured of the continuity flow around an airplane wing shape. The agreement between the theoretical calculated performance of this particular shape and the actual test in a wind tunnel was extremely close. The pressure distribution curves were almost identical. The various experiments made in the flow of water to define stream lines have all closely suggested hydrodynamic flow as have also numerous air smoke tests.

With low pressure fans pressures seldom exceed 8 in. of water or 4.6 oz. The compressibility of the air is a negligible factor. The absolute pressure of the atmosphere at sea level is 235 oz. so that any air compression would be very small, of the order of 2 per cent. The viscosity and friction of the boundaries will produce a velocity gradient different from that of the ideal fluid and for this change a reasonable allowance can be made, based on the gradient found from test.

It is necessary to bear in mind that the ideal fluid can follow any shaped passage but that air has more or less definite limitations to the degree of curvature or rapidity of change of direction allowable in a passage if the fluid is to fill completely the whole space available. Experience and good judgment must decide points of this kind. However provided the passages are so formed that the flowing air can completely fill them the flow will closely resemble the theoretical flow of the ideal fluid and on that hypothesis is based the methods of design herein described.

The application of hydrodynamic theory is treated with considerable detail by Dr. Hans Lorenz who carries his mathematical formulation into the actual design of fans. He makes an elaborate and most interesting application of Stokes's stream function. As with most purely mathematical design the difficulties of integration limit the discussion to simple formulae and prevent the free choice of the designer in matters of important proportions. Some of the Lorenz designs were actually built and tested showing fair results and giving promise of still better performance. All of his designs were, however, of the narrow wheel type usually referred to as pressure fans. The useful commercial fan is of rather the opposite type—large volume, low pressures and high r.p.m.

Bauersfeld describes an extension of the Lorenz analysis using a graphical method of integration which was used with success in the design of Francis turbine rotors. The method is long and tedious as it becomes a matter of trial and error to secure a passage that will check with certain requirements, but it is far less tedious and expensive than working on a model with a sledge and testing for better or worse.

A brief discussion of the elements of hydrodynamics and their application to Stokes's stream function may be welcome to those who may not be familiar with this subject.

In a fluid flow, limiting consideration to a flow symmetrical about an axis, there will be noted first the law of continuity; secondly, the expression, the current function $\Psi = f(r, x)$, where r signifies radial distances and x axial distances; thirdly, another function, the velocity potential, $\phi = F(r, x)$.

The law of continuity for this work may be simply stated to the effect that the mass of the fluid flowing into any closed surface must equal the mass flowing out of that surface and any set of formulae representing fluid motion must satisfy this condition. Mathematically expressed the equation is derived as follows. Consider a fluid motion having velocity components v_x and v_r in the axial and radial directions. Further consider in the region of this flow an annular ring of radius r width dx and thickness dr. The outflow from the annular space exceeds the inflow by the amount, in the axial direction,

$$\frac{\delta}{\delta x}\ (v'_x\, 2\pi\, r\, dr)\ dx$$

and radially by

$$\frac{\delta}{\delta r}\ (v'_r\, 2\pi\, r dx)\ dr$$

The sum of these two, if continuity exists is zero, that is

$$\frac{\delta\ (v_x r)}{\delta x} + \frac{\delta\ (v'_r r)}{\delta r} = o$$

This is the condition that $v'_r r dx - v'_x r dr$ may be an exact differential.

This differential can be called the Ψ function and written

$$d\Psi = v'_r r dx - v'_x r dr$$

which is the equation of the stream lines where Ψ is a parameter—constant along a given stream line. By assigning to Ψ different values the stream lines can be plotted throughout the whole region of fluid flow. The boundaries of the passage are of course made to coincide with certain selected stream lines. When the motion is irrotational, that is free from internal spin, another function ϕ, called the velocity potential is obtained. The ϕ function is such that

$$d\phi = -v'_x dx - v'_r dr$$

ϕ in this equation is also a parameter and the condition equation that ϕ may exist is

$$\frac{\delta v_r}{\delta x} - \frac{\delta v'_x}{\delta r} = 0$$

The Ψ and ϕ functions form an orthogonal system and bear the following relations to the velocity components

$$v'_r = \frac{\delta\Psi}{r dx} = -\frac{\delta\phi}{\delta r}$$

$$v'_x = -\frac{\delta\Psi}{r dr} = -\frac{\delta\phi}{\delta x}$$

A simple example of the application of these formulae is the function

$$\Psi = Ar^2 x$$

in which A is a constant. The stream lines thus represented can readily be plotted by assigning different values to Ψ. These are plotted in Fig. 1. The velocity components at any point in the region of flow are

$$v'_r = \frac{\delta\Psi}{rdx} = Ar$$

$$v'_x = -\frac{\delta\Psi}{rdr} = -2Ax$$

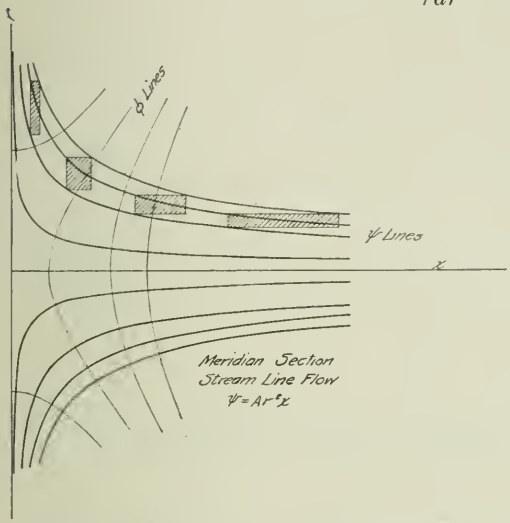

FIG. 1. MERIDIAN SECTION STREAM LINE FLOW

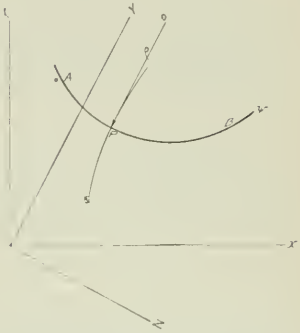

FIG. 2. GRAPHICAL ANALYSIS OF THE FLOW AT ANY POINT

Examining the values for continuity

$$\frac{\delta\,(v_x r)}{\delta x} + \frac{\delta\,(v_r r)}{\delta r} = 0$$

$$-2Ar + 2Ar = 0$$

so the function is a possible fluid motion. Further

$$\frac{\delta v_r}{\delta x} - \frac{\delta v_x}{\delta r} = 0$$

$$0 - 0 = 0$$

and the flow is irrotational possessing a velocity potential as

$$\frac{\delta\phi}{\delta x}dx + \frac{\delta\phi}{\delta r}dr = -v_x dx - v_r dr = 2Axdx - Ardr$$

an exact differential

$$d\phi = 2Axdx - Ardr$$

$$\phi = A\left(x^2 - \frac{r^2}{2}\right)$$

In this particular case the meaning of irrotational is easy to see. Note that v_r is independent of x and v_x independent of r. At any given radius therefore v_r is constant at all distances from the axis. Similarly at any given plane perpendicular to the axis v_x is constant at all radii. The result of these conditions is that a rectangle moving with the fluid would pass through the passage lengthening and narrowing but always a rectangle and always with unchanged orientation. It is the application of this function to fluid handling machines that Lorenz proposed. It is now apparent that an analysis of this kind gives complete information as to the flow of the fluid. The velocities are known at any place. The walls of the passage will coincide with any chosen stream lines.

The mathematical functions that meet the irrotational condition are few and still fewer are adaptable to the design of a fan impeller. The graphical modification suggested by Bauersfeld greatly increases the adaptability of the analysis. Consider a flow line *APB* Fig. 2 which at point P has a radius of curvature ρ. At the point P the velocity is

$$V = \sqrt{v_r^2 + v_x^2}$$

Now consider a new system of coordinates YZ having the same origin but with the Y axis parallel to P and the Z axis parallel to the direction of flow at P. The equation for irrotational flow in the new system is

$$\frac{\delta v_y}{\delta z} - \frac{\delta v_z}{\delta y} = 0$$

Now at P and for a short distance each side the osculating circle with center at 0 coincides with the stream line, and the equation of the circle is

$$y - y_o = \frac{(z - z_o)^2}{2\rho - (y - y_o)}$$

or for points close to P as $y - y_0$ is small compared to ρ it may be written

$$y - y_o = \frac{(z - z_o)^2}{2\rho}$$

Differentiating this equation with respect to time

$$\frac{dy}{dt} = \frac{z - z_o}{\rho} \frac{dz}{dt} \text{ or } v_y = \frac{z - z_o}{\rho} v_z$$

Differentiating with respect to z the following results

$$\frac{\delta v_y}{\delta z} = \frac{v_z}{\rho} + (z - z_o) \frac{\delta}{\delta z}\left(\frac{v_z}{\rho}\right)$$

At point P, $z = z_o$ and the equation becomes $\dfrac{\delta v_y}{\delta z} = \dfrac{v_z}{\rho}$

From the condition equation $\dfrac{\delta v_y}{\delta z} = \dfrac{\delta v_z}{\delta y}$ the following may be written

$$\frac{\delta v_z}{\delta y} = \frac{v_z}{\rho}$$

Further considering the velocity at P, v_z and v are identical and $dy = ds$ where s is measured on the velocity potential curve ϕ. Then $\frac{\delta v}{\delta s} = \frac{v}{\rho}$ at P.

Now if the velocity potential line is developed and from points on this as a base the velocities at those points are plotted as ordinates a curve will result such that the tangent is $\frac{dv}{ds}$. Solving for ρ above, $\rho = v\frac{ds}{dv}$ which is the sub-tangent of the curve.

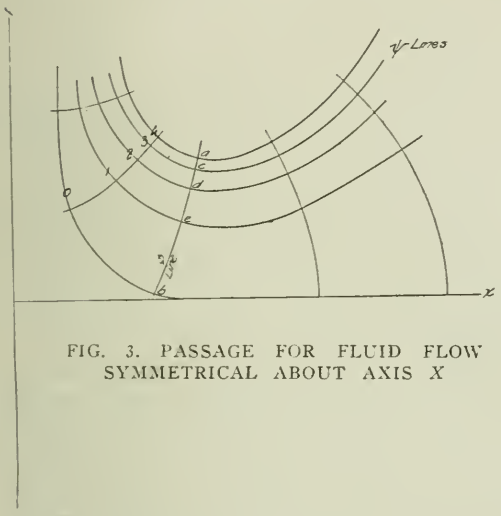

FIG. 3. PASSAGE FOR FLUID FLOW SYMMETRICAL ABOUT AXIS X

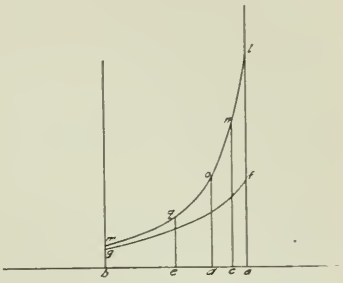

FIG. 4. DIAGRAM SHOWING HOW SCALE OF VELOCITIES AND LOCATION OF INTERMEDIATE STEAM LINES ARE FOUND

Fig. 3 shows a passage for a fluid flow with the Ψ lines, stream lines, and the ϕ lines, velocity potential lines, of a flow symmetrical about the axis x.

Fig. 4 shows a developed ϕ line and the velocity gradient gf. It is assumed that the required volume of fluid flowing through the passage is known. The velocity gradient gf has been drawn to conform to the sub-tangents which are the radii of curvature of the stream lines.

To find the scale of the velocities and to draw the intermediate stream lines proceed as follows: In Fig. 4 on the developed ϕ line as a base is drawn the upper curve whose ordinates are $2\pi rv$, assigning some convenient arbitrary scale to v.

Now as the base of this line is s developed it is apparent that the area of the diagram will be

$$\text{Area} = \int 2\pi rv\,ds$$

which is the volume flowing; the scales used for r and s are known and the correct scale for v is then readily determined after evaluating this area and equating it to the known volume.

As a further step this area is divided by ordinates into a chosen number of equal areas, four being usually sufficient. These ordinates intersect the base in the points *c,d,e*. The areas *alnc, cnod, doqe,* and *eqmb* are all equal. Now going back to the actual diagram Fig. 3 the points *c,d,e* are located on the ϕ line *ab*.

When this identical procedure has been followed for enough different ϕ lines points are found on each of these different ϕ lines which intercept equal volumes of fluid and from considerations of continuity the corresponding points lie on one and the same stream line completely delineating the flow.

The above discussion has ignored the effect of the friction of the side walls of the passage.

The flow of a viscous fluid through an actual duct is an extremely complex phenomenon, the exact nature of which is not known. Below the critical velocity the velocity distribution is, it is generally accepted, completely defined by Poisseuille's equation which gives a velocity gradient as shown in Fig. 5. In this range the pressure drop due to pipe friction varies directly as the first power of the velocity.

Osborne Reynolds first investigated the flow of fluid in pipes from the standpoint of heat transference. He found that below a certain velocity, which he called the critical velocity, the heat transfer varied as the velocity and above that velocity there was a sudden change and the heat transfer varied more nearly as the square of the velocity. He called the motion above the critical velocity turbulent.

Just what this turbulence means has never to the writer's knowledge been fully explained. If it is taken to mean a turbulent eddying motion without any stream line flow it has been shown by dimensional reasoning that the coefficient of heat transfer would increase with an increase in pipe diameter, a condition contrary to fact. Smoke tests show a stream line flow and the proved accuracy of a velocity reading with a pitot tube depends on the existence of a straight line flow in parallel lines. Prasil suggests an arbitrary function for the so-called turbulent flow that gives a velocity distribution similar to the distribution actually found by test by making the velocity a function of Ψ. And he shows that while this flow has not a velocity potential in the strict hydrodynamic sense it is a flow having a potential form. By using this approximation the result is as close to the actual flow of the real fluid as tests can show, and for design purposes satisfactorily close.

Consider now a flow in a straight pipe, Fig. 6. The gradient as determined by numerous tests is of the form shown. The Ψ function for the straight pipe flow is $\Psi = Cr^2$. The fan passage is divided into four equal volumes so that the value of Ψ for the various lines will be in the relation

$$\Psi = 0,\ 1,\ 2,\ 3,\ 4$$

A similar location of Ψ lines in the round duct will be found by dividing the pipe into two equal areas and drawing the center line. Four lines properly spaced are then shown and can be numbered 0 for the outside, 1 at the mean radius, 2 the center line, 3 the next line and 4 for the other outer wall. Then at each of these places the velocity in terms of the

average velocity can be found and using these coefficients for the correspondingly numbered Ψ lines in the actual passage the flow can be corrected for friction by simply changing the velocity gradients alone. No change will be required in the form of passage but the velocities will be somewhat different and will accordingly affect the blade shape. The flow will no longer have a velocity potential in the strict hydrodynamic sense but it will be of potential form. If desired this friction correction may

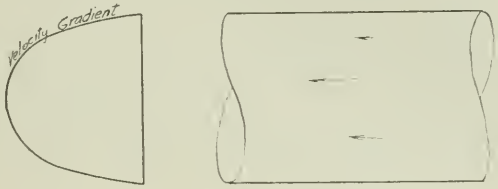

FIG. 5. VELOCITY GRADIENT TO POISSEUILLE'S EQUATION

be applied allowing for the rotational effect of the impeller hub as the rotational component can be handled independently of v'_r and v'_x. With centrifugal pumps such a procedure might be advisable but it seems unnecessary with a wide large volume fan.

In the design of a passage the procedure is as follows. First draw in the passage about as will be required. Then draw say five assumed velocity potential lines. These are developed and from the curvature of the boundaries the velocity gradients are drawn and from these the $2\pi rv$ curves. The areas are then divided into say four equal amounts and the division points on the developed bases are carried back to their places on the assumed ϕ lines.

The corresponding points are then to be connected with smooth sweeps. It will probably be found impossible so to connect the points but by translating the ϕ lines axially agreement may be secured. The curvature of the stream lines must be watched but every change after a little experience brings the solution closer.

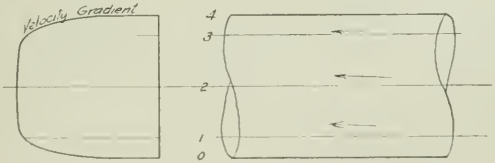

FIG. 6. VELOCITY GRADIENT FOR STRAIGHT PIPE FLOW

Having determined the shape of the main passage into and through the impeller in the above manner, equal velocity curves can be drawn if desired completely determining the velocities at any place. Any portion of the passage can be used, as the wheel making the hub conforms to the inner boundary and the side plates and inlet to the outer boundary.

The theory of blade angles used in so many works on fan and pump design based on average air velocities, peripheral speeds and angles, while useful in a qualitative manner, is most unsatisfactory for actual design.

If it were possible to have an infinite number of blades of infinitesimal thickness the usual velocity diagrams would apply, but with a finite number of blades various allowances must be made if the velocity diagram is to be of much use. Every designer has curves and coefficients for different blade angles. These may be in a form that may appear more or less reasonable and rational but really they are purely empirical data and the form and manner in which they are applied is unimportant so long as the result of the calculation is correctly related to the tests from which the coefficients were derived. It is most disconcerting to try to apply such curves and coefficients to fans of different proportions until one investigates the matter hydrodynamically and discovers the probability of the widely varying velocities. The designer's expectation then of consistency in the performance of different types based on calculations involving the average velocities of the air flow is shown to be unwarranted.

The foregoing is not meant to imply that the information of actual performance now in possession of the various fan companies is of no use. Such information is always of great value and must be relied upon in designing for specific performance as may be required in special cases. But this useful information could be compiled and utilized without resort to the velocity diagram theory. Such compilation and utilization would be equally valuable without that theory so far as quantitative determination is concerned.

The velocity diagram theory lays great stress on shock free entrance and in a geometric construction apparently secures it. Air flow and this geometric construction apparently do not agree. Even if allowance is made for the different velocities that occur across the passage at the entrance edge of the blade the maximum wheel efficiency does not occur at the volume corresponding to the determined point of freedom from shock.

Prasil suggests a hydrodynamic flow determination for the blade passages. Except in certain cases, however, to which reference will be made later the blade can be determined in a somewhat less laborious manner, namely, by applying the data secured in the experiments on aerofoils. It has been found that regardless of the shape or camber of an aerofoil that its maximum efficiency occurs when the chord is inclined from 4 to 6 deg. to the direction of the wind. This efficiency is of course the ratio of lift to drag and might not be identical with efficiency in changing the direction of the air flow, which is the requirement of a fan blade. However the efficient wing shape must be one which the air can follow without giving rise to discontinuity. This condition is also a requirement in a fan blade and it is perhaps safe to use the aerofoil experiments as a base for designing fan blades.

In the wind tunnel where the aerofoil determinations are made the flow of air is in straight parallel lines. In a fan passage the flow is diverging from the center and in curved lines. If an attempt were made to draw in a chord of the blade say at 5 deg. to the direction of the air flow difficulty would be encountered as the flow changes direction through 360 deg. Further the curved surfaces do not lend themselves to development and it would be impossible to work from projections.

However the curving diverging flow can be represented as a parallel flow on a flat surface by using a grapho-analytical method of conformal representation. The most familiar cases of conformal representation are the maps of the earth's surface. The mercator chart for example shows the lines of latitude and longitude as an orthogonal system of straight lines. The peculiarity of a conformal chart is that while the linear scale may vary from place to place all angles throughout the whole extent of the chart are the same as they are on the actual curved surface. Since only angles are of interest here the principle may be used.

Consider the curved surface of revolution A with the axis X Fig. 7 and the cylinder B with the axis z. The surface A can be represented on the cylinder B conformally as follows:

Let the infinitesimal areas $rd\theta,dl$ and $Rd\theta,dL$ be two corresponding areas. For conformity, which means that throughout the two surfaces

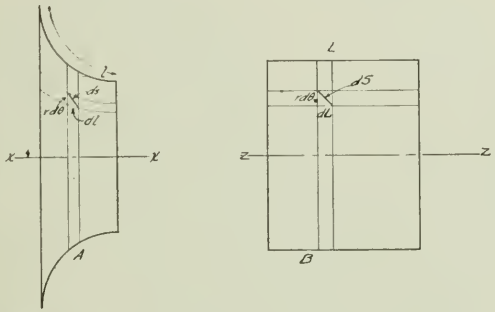

FIG. 7. CONFORMAL CHART USED IN WORKING OUT FAN BLADE SHAPES

corresponding differential areas are similar, the ratio of the diagonals of these areas must be functional and independent of the angle of the diagonal.

$$\frac{ds^2}{d\delta^2} = \frac{r^2 d\theta^2 + dl^2}{R^2 d\theta^2 + dL^2}$$

If $d\theta$ is now made the same in each case

$$\frac{ds^2}{d\delta^2} = \frac{r^2}{R^2} \frac{d\theta^2 + \dfrac{dl^2}{r^2}}{d\theta^2 + \dfrac{dL^2}{R^2}}$$

and further if $\dfrac{dl}{r} = \pm \dfrac{dL}{R}$

then $\dfrac{ds^2}{d\delta^2} = \dfrac{r^2}{R^2}$

a functional relation dependent only on the respective radii.

$\frac{dl}{r} = \pm \frac{dL}{R}$ is the condition equation.

From which $L = R \int_{r_o}^{r} \frac{dl}{r}$ is obtained.

As the generatrix of the surface A will have been determined by the graphical construction before described there will be no curve with known equation and so the formula cannot be integrated analytically.

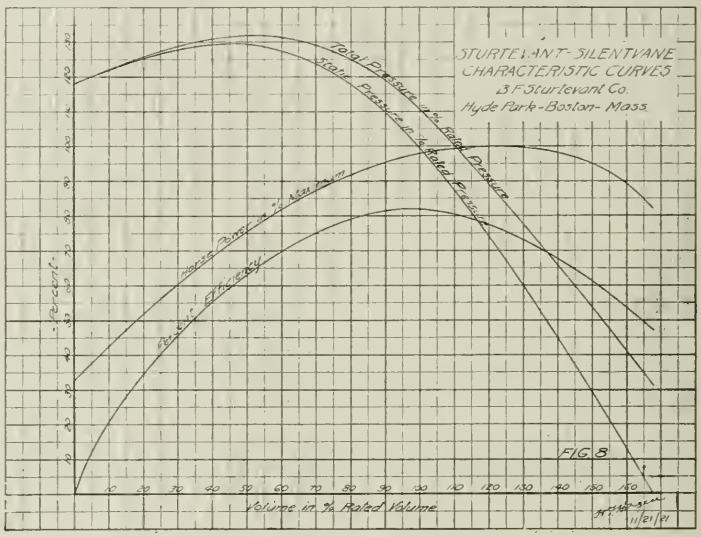

FIG. 8. CHART SHOWING PERFORMANCE OF SILENTVANE FAN

The graphical integration is secured as follows. Step off the line l in a number of small equal lengths which will be substantially straight pieces. Each of these small lengths may be considered a Δl and with the mean radius of the respective pieces $\frac{\Delta l}{r}$ can be calculated. These values are then summed up to get the value of $\int \frac{dl}{r}$ for each piece by adding successively the next value to the sum of those preceding. From these values L can be calculated. Then as given above, as $\theta = \theta$ for both cases, any curve or line on the warped surface can be charted on the cylindrical surface or the cylindrical surface developed. Now on this developed surface the lines of air flow relative to the earth are straight

parallel lines, elements of the cylinder, for the condition of no work; that is when the air is passing through a blade free fan passage.

From previous determination of the actual velocity gradients through the fan the relative path of the air for a given fan speed can be drawn and brought to the conformal chart. A line 5 deg. to the elements of the cylinder can now be laid out on the conformal chart that is to the direction of the wind and this line can be used as the chord of the blade. Then on this chord the effective blade curve can be drawn, determining its actual shape as requirements may demand and as good judgment and experience suggest. This curve will be the absolute path of a particle moving along the blade. The relative path or actual chart of the real

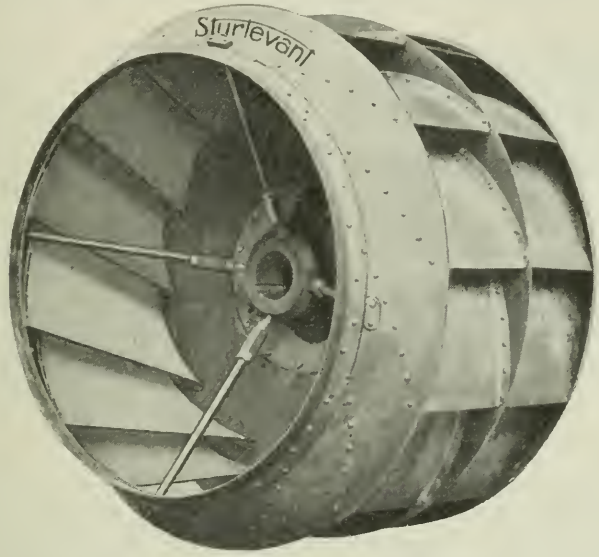

FIG. 9. SILENTVANE FAN DESIGNED ACCORDING TO METHODS OUTLINED

blade is readily determined from the absolute and relative paths of the work free condition. Carrying this charted curve of the blade back to the actual surface of the fan passage gives the required blade shape. For blades of large changes in angle between entrance and exit this method will not serve, but a calculation of the hydrodynamic flow in the relative channel will be necessary. However such sharply curved blades can never develop the efficiency of the gently curved type. For the more involved calculation the conformal chart again serves well, enabling all the work to be done on a flat surface. It is necessary of course to make corrections in velocity in accordance with the functional relation between real surface and chart.

The methods described herein were used throughout the design of the fan shown in the cut Fig. 9. In this fan there is not a single arbitrary curve. The main passage was determined to conform to the laws of irrotational flow, due allowance was made for friction, the blades laid out having effective chords at 5 degrees to the air flow. The performance

of the fan is shown on the accompanying chart, Fig. 8. The high efficiency—higher it is believed than ever before reached in fan design; the large volume for a given wheel diameter and the freedom from objectionable noise—noise in a fan is an indication of poor efficiency—all go to show that the theory is headed in the proper direction. Incidentally other desirable features have been secured including high r.p.m. for direct motor connection and a self limiting horsepower characteristic.

The extraordinary results given by this fan—the first fan to better the efficiency of the old paddle wheel—are due in the writer's opinion entirely to the consideration given in shaping the passages to conform to a natural air flow in accordance with the laws of theoretical hydrodynamics.

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No. 628

COMPARATIVE TESTS OF AUTOMATIC VENTILATORS

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Non-Members

BECAUSE of the lack of authentic data upon the effectiveness of different types of automatic ventilators, when subjected to varying conditions, the tests, described in this paper, were conducted in the Engineering Experiment Station of the Kansas State Agricultural College, to determine the efficiency of many of the various styles.

A preliminary series of tests upon three different types of automatic ventilators were carried on at the Kansas State Agricultural College during 1919 and 1920. The results of this investigation proved so interesting that a decision was made to continue the work upon a much broader scale and to include as many of the commercial ventilators as possible. Consequently the laboratory equipment was enlarged and twenty-two representative ventilators were secured through the courtesy of the various manufacturers.

TYPES OF VENTILATORS

In general, an automatic ventilator is simply a protecting device, which is placed over a hole in the roof of the building or inclosure to be ventilated, to prevent the entrance of rain or snow. Their use for the removal of foul gases from inclosures is quite common so that further description is unnecessary. Their chief advantage over the mechanical means of ventilation consists in their providing ventilation without the use of mechanical power.

Such a ventilator, in order to be effective in the production of ventilation, should not only be effective in preventing the entrance of snow or rain, but should also provide for the free exit of foul gases when no wind is blowing and should not be affected by down drafts when winds prevail. This fact gives rise to various types, the main feature in their several designs being to utilize the wind to better advantage in the production of draft. Commercial ventilators may, consequently, be divided into four general classes dependent upon the means utilized in producing the additional draft. They are the plain stationary, the siphoning stationary, the plain rotary, and the rotary siphoning types.

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Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

The plain stationary type of automatic ventilator makes no special provision for utilizing the wind velocity in producing additional draft. Examples of such ventilators are illustrated in Fig. 1. This type consists simply of a cap over the ventilator pipe which prevents the direct entrance of rain or snow and is made storm proof by a circular cowl or hood.

FIG. 1. TYPES OF AUTOMATIC VENTILATORS CLASSIFIED AS PLAIN STATIONARY

The stationary siphoning type is illustrated in Fig. 2 and its principle of operation is based upon the breaking up of the wind currents and directing them in such a manner as to create a decreased pressure in the upper portion of the ventilator. Ventilation is thus secured by an ejector action. When no wind is blowing, the ventilation resulting is that due to natural circulation of the air. When winds prevail the siphoning action is established and additional ventilation results. The higher the velocity of the wind, the greater the exhausting power of the ventilator.

The plain rotary type, Fig. 3, consists of simply an elbow or its equivalent which is supported upon a vertical shaft. The position of the elbow is regulated upon the principle of the weathervane so that the opening from the ventilator always points away from the direction of the wind. This ventilator makes use of the slight vacuum produced by the wind in the production of additional draft.

The turbine ventilator, Fig. 4, was classed in this investigation as of the rotary siphoning type in that its operation was difficult to classify and that its effectiveness brought it within the rotary siphoning type. It is illustrative of the many different ideas which have been incorporated in ventilator construction.

FIG. 2. TYPES OF VENTILATORS CLASSIFIED AS STATIONARY SIPHONING

The rotary siphoning type is shown in Fig. 5. It has, in addition to the principle embodied in the plain rotary type, a feature similar to that used in the stationary siphoning ventilators. The air is directed by flutes or vanes so that an ejector action is established, thereby increasing the

velocity through the ventilator. In some cases, the ejector is placed within the ventilator, while in others it surrounds the ventilator.

METHOD OF TEST

In establishing a method of procedure in conducting the tests, it was conceded that there were many factors which would influence the practical performance of a ventilator and which should be included if an exhaustive test were to be conducted, but difficulties would arise if any attempts were made to include these in the laboratory tests.

In dealing in a practical way with the effectiveness of a ventilator, it is necessary to take into account the openness of the structure to be ventilated, as well as the action of the wind. When wind is arrested by an obstruction, such as a building, a pressure is developed which forces air

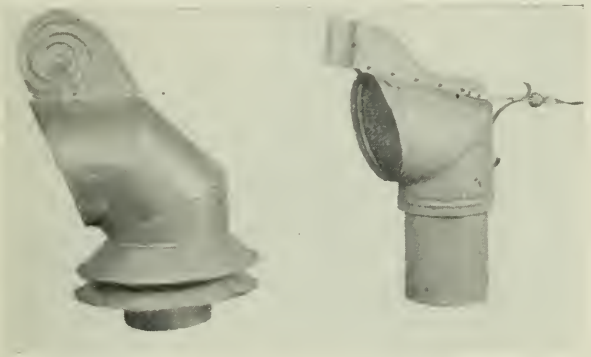

FIG. 3. TYPES OF AUTOMATIC VENTILATORS CLASSIFIED AS PLAIN ROTARY

through possible openings with the result that the air inside the building is forced outward through no action of the ventilator other than the opening it provides.

Furthermore, temperature differences exist between the inside and outside of buildings ventilated and this likewise will increase or decrease the effectiveness of a ventilator in practical use, depending upon the degree of temperature difference.

In conducting the tests in this investigation, it was finally decided that the performance of the wind in inducing a current of air through the ventilator was the important factor. Also, it was thought desirable to secure data upon the advantages of the various types of ventilators and to ascertain, if possible, what principles of design should be incorporated to secure the the most efficient ventilator. The investigation, consequently, was limited to these phases of the subject.

TEST EQUIPMENT

In order to conduct the tests and to approximate actual wind conditions, a wind tunnel 3 ft. square and 16 ft. long was first constructed. In one end of the tunnel a variable speed fan was placed which was capable of producing wind velocities up to 15 miles per hour. About 3½ ft.

from the other end of the tunnel a 10 in. pipe, which protruded about 2 in. above the tunnel floor, was inserted to receive the ventilator.

The wind tunnel with its fan was placed in the mechanical engineering laboratory, the volume of which was large enough to eliminate all perceptible drafts caused by the fan. The temperatures of the air in the tunnel and that in the room were also equal so that all effects from natural ventilation were removed. Honeycombed screens were placed within the tunnel to produce parallel currents of air and to create a more uniform velocity over the tunnel section.

All the ventilators tested were of the same commercial size. The diameter at the base of the ventilator was 10 in. This dimension was selected as being fairly representative of the ventilators used in practice and this size made possible the investigation without necessitating a large tunnel. It was assumed that the manufacturers of ventilators proportioned their ventilators according to their size and that a 10-inch ventilator would be proportional to other sizes.

FIG. 4. TURBINE VENTILATOR CLASSIFIED AS A ROTARY SIPHONING

In performing the tests, a ventilator was placed over the pipe in the tunnel and the speed of the fan regulated to produce the desired wind velocity. The air from the fan was forced through the tunnel toward the ventilator, thus creating a region of high pressure near the ventilator which tended to force air down through the ventilator rather than induce a current up through it. Readings of the wind velocities in the tunnel and of the air induced through the ventilator were taken by means of an anemometer. The readings of the velocity of the air in the tunnel were taken at various sections of the tunnel and the results averaged. The velocity of the air induced through the ventilator was measured by inserting the anemometer in the short vertical pipe underneath the tunnel floor. Data was also secured of the air induced through the ventilator pipe when no ventilator was inserted in the tunnel. The results secured in this latter case are referred to and recorded as *No Ventilator*.

CONCLUSION

Before attempting to draw any conclusion from these results, it may be well to call attention to one or two important facts.

1. The test apparatus while designed to reproduce actual wind conditions approaches this only approximately. As was mentioned before, in an actual installation, the wind has its progress arrested by the building. A portion of this air

enters the building through any crevices. This theoretically increases the pressure within the enclosure and materially increases the velocity of the air passing through the ventilator. Thus, it would seem that the results secured in this investigation were only approximations of actual installations or would be applicable only to exceedingly tight or well built buildings.

To compare the results with actual installations, a 10 in. ventilator pipe was inserted in a small outbuilding. No heat was supplied to the building during the tests and readings of the wind velocity and the induced velocity through the ventilator were read when various types of ventilators were used. The variable velocity of the wind made the readings rather erratic, but when the tests extended over several days, the average results checked fairly closely with those secured in the laboratory. It was thus concluded that the laboratory results were fairly representative. They were at least conservative values of what should be expected in actual installations.

2. The results were secured on rather small ventilators. Commercial ventilators as a rule are much larger than those used in this investigation and their

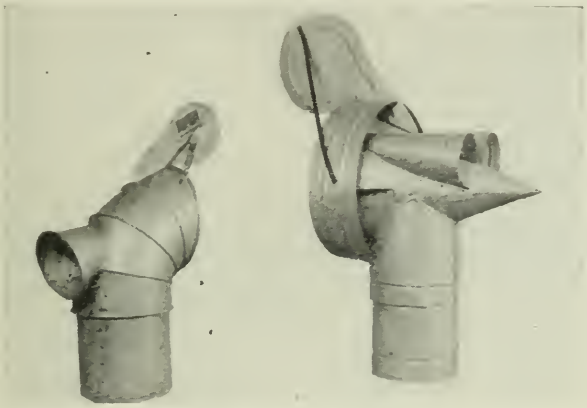

FIG. 5. TYPES OF AUTOMATIC VENTILATORS CLASSIFIED AS ROTARY SIPHONING

proportion may be different from that of the 10 in. ventilator tested. This may alter the aspect of the problem, although it is assumed to no serious extent.

3. The results as found in the investigation differed somewhat from those secured in tests of similar ventilators conducted in different laboratories. This discrepancy is partly explained by the fact that different laboratory equipment will produce different results. Any turns or extension to the pipe which is used for receiving the ventilator under test produces friction that lowers the test results. The size of the tunnel likewise plays an important part, a small tunnel producing different results from those of a large one. With a tunnel 3 ft. square as used in this investigation, some of the larger ventilators produced so much obstruction that only a comparatively small area remained for the flow of air between the ventilator and the sides of the tunnel.

One of the first important conclusions that may be readily drawn from these results is that a wide range of effectiveness is found in the ventilators of each type. There were ventilators in each type that gave an extremely high ventilating effect, while others were comparatively poor. The range was more pronounced in the plain stationary type. Some of the plain stationary types did not give as good results as no ventilator. Much of this could be overcome through more judicious design. The free area through the ventilator openings should be of ample capacity

TABLE 1. RESULTS OF TESTS SHOWING THE VELOCITY IN FEET PER MINUTE INDUCED THROUGH SEVEN DIFFERENT PLAIN STATIONARY VENTILATORS

Velocity of wind in miles per hour	No. Ventilator	Velocity induced through ventilator, feet per minute Ventilator Designation 1	2	3	4	5	6	7
4	145	185	266	148	185	142	133	160
8	241	287	355	257	287	238	242	267
12	337	390	446	366	390	334	350	375

TABLE 2. RESULTS OF TESTS SHOWING THE VELOCITY IN FEET PER MINUTE INDUCED THROUGH SIX STATIONARY SIPHONING VENTILATORS

Velocity of wind in miles per hour	No. Ventilator	Velocity induced through ventilator, feet per minute Ventilator Designation 1	2	3	4	5	6
4	145	162	157	226	189	205	206
8	241	304	292	404	315	332	369
12	337	446	426	583	440	458	532

TABLE 3. RESULTS OF TESTS SHOWING THE VELOCITY IN FEET PER MINUTE INDUCED THROUGH FOUR PLAIN ROTARY VENTILATORS

Velocity of wind in miles per hour	No. Ventilator	Velocity induced through ventilator, feet per minute Ventilator Designation 1	2	3	4
4	145	208	192	191	202
8	241	346	348	354	341
12	337	484	505	518	480

TABLE 4. RESULTS OF TESTS SHOWING THE VELOCITY IN FEET PER MINUTE INDUCED THROUGH FIVE ROTARY SIPHONING VENTILATORS

Velocity of wind in miles per hour	No. Ventilator	Velocity induced through ventilator, feet per minute Ventilator Designation 1	2	3	4	5
4	145	257	192	222	217	204
8	241	479	370	387	410	393
12	337	702	548	553	606	582

TABLE 5. RESULTS OF TESTS SHOWING THE AVERAGE VELOCITY IN FEET PER MINUTE INDUCED THROUGH THE VARIOUS TYPES OF VENTILATORS

Velocity of wind in miles per hour	No. Ventilator	Velocity induced through ventilator, feet per minute Type of Ventilator Plain Stationary	Siphoning Stationary	Plain Rotary	Rotary Siphoning
4	145	168	191	198	218
8	241	273	326	348	408
12	337	379	461	497	598

and be as free from obstructions as possible. The storm band, if used, should be of ample width to prevent the wind from entering the ventilator.

As judged from the results of the different types of ventilators tested, a slight gain is made by utilizing the wind to better advantage. In the order of their effectiveness come the plain stationary, the stationary siphoning, the plain rotary and the rotary siphoning. The average effectiveness of the various types is shown graphically in Fig. 6.

A comparison of the siphoning types of ventilators with those of the non-siphoning types show that some of the non-siphoning types are as effective as some of those employing the siphoning principle. This fact led to a special investigation which had for its object the determination

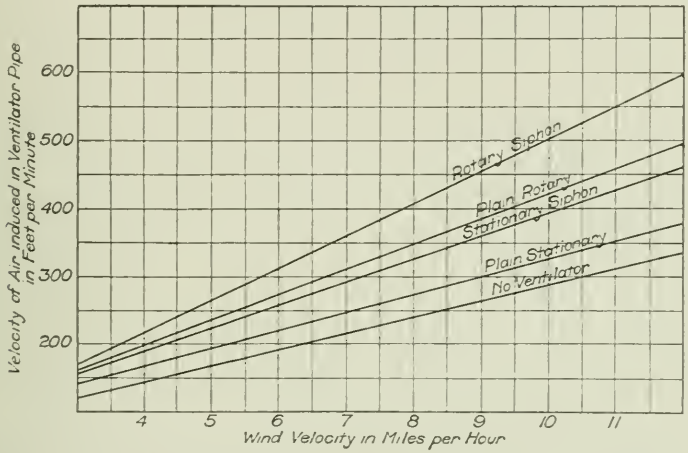

FIG. 6. THE AVERAGE EFFECTIVENESS OF THE FOUR TYPES OF VENTILATORS TESTED

of the effectiveness of the siphons. In this part of the work, the siphons on the windward side of the ventilator were closed by wads of paper and invariably the effectiveness was not decreased.

A further study of the effectiveness of the stationary siphoning types was made to determine what factors in the design made those of one manufacturer better than another. This study was decided upon because most of the ventilators in this classification were practically the same in general measurements. It was found that the width of the storm band played an important part, the wider the band, the greater the effectiveness.

The results also seemed to show that the addition of the flutes for producing the siphoning acted as a secondary storm band and that those ventilators in which the angle of the flutes were small proved the most effective. This later conclusion was explained by the fact that with ventilators whose flutes were steep, more difficulty was encountered by the air in entering the ventilator. Consequently, less air entered that had to be exhausted and the exhausting power of the ventilator could be utilized in inducing air through the ventilator rather than removing air that had been introduced for siphoning.

The results of this investigation seem to indicate that the most effective action of the wind in inducing air through a ventilator is the vacuum produced in the wake of the wind. The ventilators which showed marked effectiveness in these tests took advantage of this principle. Those ventilators, which presented a large obstruction to the wind, other factors being the same, gave best results.

The factors which seem to be important in the design of a ventilator are as follows: The area for the outgoing gases should be made ample; in the case of siphoning ventilators the free area should be designed to care for the additional air used in siphoning; the storm band, if used, should be made at least wide enough to prevent entrance of outside air, any provision, whereby the vacuum created by the wind is increased or made more effective, will produce better results.

EFFECT OF STORM BAND

In order to test the effect of the width of the storm band, an experimental ventilator of the plain stationary type was constructed. The storm band was formed by two sheets of tin and made so that the width of the band above and below the ventilator opening could be varied. The size of the tunnel did not permit of a larger storm band than 22 in. and the minimum width of band was 17½ in. The lower edge of the storm band was placed 4 in. below the ventilator opening as this location gave best results. Table 6 gives the results of this test. A comparison of these results indicates that a wide storm band materially increases the effectiveness of the ventilator and that a plain stationary ventilator so equipped is equal in effectiveness to some of those of the rotary siphoning type. While these results may be in error because a ventilator with so great a band width produced so large an obstruction in the experimental tunnel, they do indicate the advisability of considering the storm band as an important part of the ventilator.

TABLE 6. RESULTS OF TESTS UPON AN EXPERIMENTAL VENTILATOR TO DETERMINE EFFECTIVE WIDTH OF STORM BAND

Wind Velocity in miles per hour	Velocity induced through ventilator, feet per minute Width of Storm Band in inches			
	17½	19	20	21
4	130	164	183	202
8	402	432	456	477
12	620	651	680	696

DISCUSSION

HARRY M. HART: Mr. President, it might be of interest to know that some of us conducted some tests on the various types of ventilators in Chicago some years ago, and as I recall it we tested just an open pipe without any hood on it, and we got better results, as far as efficiency is concerned, from the open pipe without the ventilator on it than we did from any type of ventilator. Of course there was no storm hood. But it gave you a principle on which to work in designing ventilator hoods. The tests mentioned were conducted by the Chicago Commission on Ventilation and have been published.

H. F. Hagen: I think it has been definitely proven by a number of tests—I know we have in our own laboratory—that a straight pipe will not produce as big a suction or as big a velocity in the draft tube as will certain of the types of ventilators. There are some ventilators that will not do as well as a straight pipe. But most of the modern ventilators that you would be apt to use, at least, the rotary ones, will produce better result than a straight pipe, and in addition give you storm protection. The only trouble with designing wind ventilators is that the wind is there and the actual efficiency of the ventilator has very little bearing on it.

I think it might be shown, however, that for a given amount of money you can get about as much from the ordinary cowl as you can get with any of the ordinary ventilators. The cowl would have to be larger, it is true, than some of the better types of ventilators, but except in very few instances the size of the pipe is not the limiting feature. If you can simply put on a larger pipe and put on the ordinary cowl ventilator you will do about as well per dollar as you would with the more elaborate equipment.

This is the great objection, the great difficulty, the great obstacle that confronts a designer when he starts in to design a ventilator. Ventilators that would produce very much better results than any of these could easily be designed, but the cost would run up to such a point that it would hardly be worth while.

President McColl: Some of us may remember a paper read at our Philadelphia meeting—I think it was by a man from the U. S. Bureau of Standards—on testing ventilators, in which in a series of tests they took the open pipe as a standard, 100 per cent. The rotary and various makes of ventilators showed a great many, at least, below the 100 per cent standard and a great many above the 100 per cent; indicating that some ventilators are less effective than the open pipe and some are more effective.

No. 629

CORROSION AND ITS PREVENTION IN VESSELS CARRYING WATER

By J. R. McDermet, Jeannette, Pa.[1]
Non-Member

CORROSION is generally considered a non-preventable waste, but much good toward its prevention may be accomplished in certain fields, to which it is intended to confine this paper. The subject, as announced, indicates the scope as *vessels carrying water*. In an engineering sense, these vessels may be either water piping, boiler economizers, boilers, steam piping or water tanks. These classifications, necessarily do not cover all cases of corrosion of metal by water. They do, however, cover the bulk of that which is preventable.

In the case of pipes different life factors may be expected in different localities, but the ultimate life ordinarily can be foretold and it may be either more or less, according to the locality, than the useful life of a building. The man handling power boilers, recognizes the advantages of economizer units but is deterred from their adoption by the high depreciation and maintenance cost due to unchecked corrosion. Boiler corrosion presents anomalous features. Ordinarily, steam pipes would not be considered in the category of vessels carrying water, but in central station steam heating service, saturated steam is usually encountered, together with a certain quantity of moisture and where moisture appears in the steam, corrosion usually accompanies it.

It is possible to say, non-technically, perhaps, but still with a high degree of accuracy, that corrosion results where water, oxygen and a metal are present and the metal is not a noble metal. If either the oxygen or the water be removed from the combination, corrosion, from a practical and observable standpoint, ceases.

Iron has a natural and fundamental tendency to dissolve in water. This tendency is measured by its solution pressure which may be expressed in terms of electric energy. Iron, therefore, in contact with water, tends to dissolve, but the quantity going into solution is so minute that it is not observable. In the process of solution, the solution potential of the iron relative to the water is changed, so that the electric forces originally present, and tending to drive the metal into solution, automatically diminish as the solution increases. There is, therefore, a very natural and positive limit to the solution of iron in water. This solution

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may be attained initially by the water in circulation through earth strata or it may be attained by dissolving from pipes. In any event, over a lifetime, it does not represent an appreciable quantity.

If, however, oxygen is present in the solution, the natural checks which are provided for limiting the solution of the metal in water are displaced and equilibrium conditions are impossible of attainment. Iron going into solution in water displaces ionic hydrogen from the water, and the ionic hydrogen collects on the metal surfaces and creates an electric potential opposing the tendency of the metal to dissolve. This hydrogen potential is merely an inhibitive. It has no relation to the ultimate value of metal which will dissolve, but it may vastly increase the time during which solution is taking place. Oxygen in solution, by combining with this polarizing hydrogen to form water, depolarizes the metal as an electrode and accelerates the reaction of metal and water without, however, changing the ultimate value of metals which will dissolve.

As the metal has a solution pressure relative to the water, which may be measured in terms of electric potential, similarly, the solution, after some iron has been dissolved in it, has a similar electric potential, operating in the reverse direction and of different polarity, which tends to deposit metal out of solution upon the electrode. Equilibrium is necessarily attained therefore, when the metal dissolving from the electrode is equal to the metal deposited upon it by the solution. The quantity of metal necessary to be dissolved to provide the potential, is almost infinitesimally small.

The presence of oxygen and its ability to combine with other elements or substances frequently interferes with the natural check of counter potentials, which nature provides to limit the corrosive process, often destroying it as fast as it is created. The corrosion is not only accelerated, but it is simultaneously freed from all restrictions and allowed to continue uninterruptedly until either the iron or the oxygen is consumed. It is for this reason that oxygen is ordinarily considered as the corroding agent. Its function is, of course, to disturb the natural causes which limit corrosion and allow it to continue uninterruptedly.

Certain types of dissolved solids, notably magnesium salts, which form insoluble hydroxides and liberate acid forming ions in the water, exhibit a regenerating action with oxygen which not only accelerates corrosion due to acidity of the water, but maintains the acidity irrespective of the amount of corrosion which has previously taken place, there being no exhaustion.

The electric solution potential of a metal to water is not only a function of the character of the metal and water temperature, but also of the dissolved solids, which are present in the water in ionic form. As a result, pipe lines carrying natural water, exhibit different degrees of corrosion in different localities. It is generally true that waters having low chloride content and high calcium are least corrosive.

It appears, as far as the expectation of useful life is concerned, that iron and steel pipes are on a parity. This is not altogether consistent with an existing opinion that wrought-iron pipe exhibits greater resistance to corrosion than steel pipe. Commercial wrought-iron pipe of

today still exhibits the fibrous structure of wrought iron, but from a corrosion standpoint, its characteristics are not those which are usually associated with wrought iron. It is questionable, therefore, whether the present liability to corrosion is due to poor grades of so-called wrought iron or whether the original opinion ascribing immunity to it was not in error. The characteristics of the two metals, as far as corrosion appearance is concerned, are quite distinct since the pitting of the steel pipe exhibits a uniform allocation while that of the wrought iron closely follows the fibrous structure.

In the localities where corrosion is virulent, water standing in pipes for any appreciable length of time, speedily becomes discolored with red rust. This red rust is not only soap destroying from the standpoint of cleansing purposes, but it also is distasteful to the user and has a high staining effect upon porcelain fixtures as well as upon fabrics in laundering processes. In itself, it constitutes a very inconsiderable quantity of iron, since the products of corrosion are necessarily much greater than the amount of metal participating in them, but it has a high coloration.

Not all of the products of corrosion remain suspended in the water. Ordinarily, the greater portion of them, adhere to the interior of the pipe. An analysis of the scale or corrosion products ordinarily indicates perhaps 15 per cent to be insoluble calcium compounds, which are mechanically mixed with the various oxides of iron. As a result, in severe cases of pipe corrosion, unless the velocities are higher than those usually encountered in domestic service, the pipes become either totally or partially stopped.

The corrosion products, clinging to the interior wall of the pipe, exhibit several layers of stratification, depending upon the degree of oxidation of the oxide involved, the ferrous oxide, which has the greater amount of iron, proportionally to the amount of oxygen, being immediately adjacent to the pipe surface. It was explained previously how oxygen is available in oxidizing soluble products of iron. When insoluble corrosion products occur in a pipe, the reverse action is also true if the water is completely deoxygenated, and the insoluble products of iron oxide are reduced from the ferric to the ferrous and become again soluble, but at a very slow rate. Part of the corrosion products become truly dissolved in the water in the ferrous state and are not noticeable. In the reduction process, other portions and in particular, the insoluble calcium residue becomes detached and collects in the low parts of the piping system. Ultimately, all but the calcium compounds become dissolved.

Ordinarily, the failure of a section of pipe is through perforation, produced by a pit hole, probably originating, initially, in some surface imperfection on the interior of the pipes and when they once form, the corrosive conditions are aggravated by their presence. It is, therefore, simply a question of relative time until some place in the pipe is perforated. Ordinarily, very little warning is given on the exterior, indicating a probable failure of the pipe.

The average life of iron pipe in the worst localities is 5 years. In other localities, it may extend to as much as 15 years and available records indicate in some few specific instances that pipe lines in the Great Lakes

region have shown a life as high as 18 years. Ordinarily, the most accelerated conditions of corrosion occur, of course, with high water temperatures and it is economically feasible only to protect hot-water systems. Cold-water systems, however, are not immune and as a result, the question of corrosion of hot-water systems is intimately connected with the question of the temperature at which they operate. It is, therefore, impossible to generalize extensively upon expectation in the matter of length of pipe life. It is, however, possible to say that the average useful life of a building exceeds the life of the pipe.

Brass pipe has been, in many instances, installed in view of the realization of the limitations of iron pipe. Unfortunately, it only constitutes a makeshift. It is the expectation generally that brass pipe will have a life slightly greater than twice the life of iron, but the phenomena are very complicated. Brass pipe, is unfortunately, not a uniform product and it, therefore, shows in many cases most unusual behavior. In some cases, brass pipe life has been less than iron.

Ordinarily, the failure of brass is from dezincification and naturally, dezincification rests somewhat upon the initial zinc content of the brass as well as its preliminary treatment. Brass pipe is more sensitive to impurities in the water than is iron, there being less resistance to acid waters as well as water polluted with nitrogen compounds, either from sewerage or from decaying vegetation. Slight manganese content is also extremely disastrous.

In the corrosion of iron, attention is primarily directed always to the relative acidity of the water and the dissolved oxygen content. In the case of brass, the observation necessarily needs to be extended to carbon dioxide, since zinc is particularly susceptible to attack from carbon dioxide. The process of deaeration, therefore, must include not only the permanent gases of the air, but also carbon dioxide, which is at best, an uncertain quantity.

Previous mention was made of dezincification as the responsible cause of corrosion in brass pipe. Primarily, from a physical observation standpoint, the corrosion behavior of brass is not pitting or perforation, but embrittlement. In the process of leeching out the zinc from the alloy, the remaining constituents are left in an extremely brittle state, very similar to their condition when they are deposited under a high current density upon an electrode from a plating solution. They have the evidence of being saturated with hydrogen. The further significant condition arises that dezincification is apparently accelerated by mechanical strains and failures of brass pipe almost invariably occur at the threads, which was initially a cause of suspicion in the erection of the pipe at the beginning. Temperature expansion strains bring the incipient thread failures to light and if water temperatures on a hot water system of brass vary rapidly and over large ranges, it is to be expected that this pipe will show a much lower useful life than if temperatures are maintained constant.

Steel tube economizers exhibit roughly, the same corrosion phenomena as pipe, except that the corrosion is ordinarily concentrated upon pit hole localities rather than distributed as widely as in pipe. Cast iron economizers ordinarily show wasting, and are particularly susceptible to

corrosion with high water velocities and very pure water. They have naturally, a much higher resistance to corrosion than steel but are open to suspicion on account of low factors of mechanical safety and have a very limited life where cleaning is necessitated.

Cast iron initially, when first exposed to water and oxygen in solution, has a much higher liability to corrosion than steel. The corrosion products, however, include a certain amount of silica and graphite, which, if undisturbed, provides a truly protective coating from the iron and arrests corrosion. If, however, the protective coating is for any reason disturbed, either by erosion or by mechanical cleaning, the corrosion of the cast iron proceeds most rapidly. It appears, also to be more sensitive than steel to water polluted with nitrogen compounds either from decaying vegetation or sewerage.

It may ordinarily be assumed that corrosion is more severe as temperature is increased for a given initial saturation of dissolved oxygen. However, in the operation of power plants, heating of the water eliminates, to a fair extent, the oxygen dissolved and, as a result, economizers have invariably shown a longer useful life at high temperatures than at low and it is in a very great many cases, perfectly feasible to operate a cast-iron economizer at high temperatures without any attempt at deaeration, except very liberal venting of steam from open feed water heaters. It does not seem economically safe to operate steel tube economizers without some means of deaeration. The useful effect of an economizer and its efficiency is not only a function of the flue gas, but also a function of the water temperature and the economizer effect is much greater with lower water temperatures, which in most cases, may be prohibitive without some means of deaeration. Apart from the presence of moisture, dissolved gases in the feed water appear to have no effect upon the steam lines. Where condensate in central station heating plants is collected and returned, the condensed waters are extremely virulent in their corrosion. The return condensate pipes ordinarily do not run full of water, but carry a trickle along their lower side and failures on condensate lines almost invariably occur in horizontal runs of pipe and from channeling on the interior, the manifestation being a little furrow or ditch, where the stream of water has followed the bottom of the pipe.

Boilers exhibit no uniform characteristics of corrosion. The writer has previously presented, before a meeting of the *American Society of Mechanical Engineers,* a paper which embodies his best experience in interpreting boiler water analysis, but it is unfortunately impossible to condense the conclusions in the brief space which is here permissible. It is desirable to say, however, that corrosion in boilers occurs either at the water line or in dead circulation spaces in boilers which are not operated at extremely high rating. It is probably true, however, that corrosion from dissolved gases is not a serious factor in boilers operating on natural water. If any scaling is present, the scale, while retarding the heat flow, also serves as a protective coating against corrosive action.

An attempt has been made to indicate the causes and the manifestations by which corrosion may be observed. It remains, therefore, as against its previous category, to suggest possible remedies. One of the

first and simplest is alkalinity of the water. The solution potential of iron, which is the initial incipient cause of all corrosion, varies with the chemical character of the water and particularly with the concentration of hydrogen ion. It is possible, therefore, experimentally to establish a solution density of hydrogen ion at which there will be no measurable tendency of the iron to dissolve. Unfortunately, this concentration is expensive when dealing with huge quantities of water, and it is also extremely difficult if not impossible to maintain in boilers and economizers. It is of a characteristic soapy taste which prohibits it in hot water domestic service. It is, therefore, a simple remedy but not a useful one.

Certain brands of paint are available, which are very effective for a limited time, when properly applied. If the surfaces are such as exist in tanks where they may be painted at proper intervals, this protection is more than sufficient. Unfortunately, pipe lines, economizers, etc., are not accessible for the application of paint at the frequent intervals which are necessary.

The third and perhaps the most generally useful method is deaeration of the water which aims at the correction of the fundamental cause and automatically uses the inhibitive effects of alkalinity as far as they are practically available.

Several methods are commercially available for accomplishing either complete or partial deaeration and each has different degrees of flexibility and produces correspondingly different ranges of deaeration. It is within the writer's province to speak authoritatively, if at all, only upon the Elliott process, which uses explosive boiling.

Doubtless it is generally agreed that the simplest way of eliminating dissolved gases from water is to reduce the solubility to zero, if this can be accomplished in a practical fashion. It is this feature upon which the Elliott process of deaeration is based, and the methods and apparatus are intended to accomplish this result and at the same time, render the process 100 per cent efficient in heat recovery.

Reference to a solubility curve for air-gases in water, will show that ordinarily these curves are plotted as functions of solubility, and temperature of solvent. The one significant fact, however, remains, namely —that any curve so plotted has still a third variable which for the conditions of plotting is constant, namely—the partial pressure of the solute. Solubility curves, therefore, are ordinarily given for certain arbitrary pressures, and the usual pressures are either a total air pressure over the entire range of temperature equal to the pressure of the atmosphere or a total pressure of air and water vapor equal to the pressure of the atmosphere. Since solubility is invariably attained from atmospheric mixtures, the partial pressures of the constituents are readily deducible, oxygen for instance, being approximately 20 per cent of the total.

Further reference to a solubility curve for atmospheric air, in which the sum of the water vapor pressure plus the pressure of the partial air component is equal to 29.92 in. of mercury, shows that this curve approaches zero at 212 deg., which is the temperature of boiling. Necessarily, at this pressure, the partial air component is zero, since the vapor pressure of the water by definition is equal to 29.92 in. of mercury at the boiling point. The Elliott process by lowering the pressure of the

region in which the boiling occurs automatically moves the boiling point, and as a result the point of zero solubility over the entire range of useful temperatures.

Boiling, then, is an efficient way of removing dissolved gases from water. Unfortunately, boiling at atmospheric pressure requires a heat source extravagant from the standpoint of heat recovery, and also an excessive boiling loss. Vapor boiling under reduced pressures makes utilizable normal exhaust steam temperatures as heat supply. If some means then are added to permit the heat recovery from the boiling process, the apparatus becomes efficient and utilizable. This feature is the ideal upon which the Elliott process is based.

While it is possible to boil air out of water, as everyone knows, it is nevertheless difficult to do it efficiently, and boiling even under atmospheric pressure follows certain definite laws of heat quantities and rate of heat addition. In the Elliott process, heat is added to the liquid under pressure conditions at which the boiling point is not attained. This water heated to a predetermined temperature below its boiling point is then suddenly admitted into a region of vacuum, this vacuum being predetermined in consideration of the initial temperature of admission and the temperature desired for using. When the water enters this region of vacuum, it is superheated and its immediate tendency is to boil with explosive violence at the expense of the heat of the liquid, which was previously added and is now available under the altered conditions of pressure. This explosive boiling results in a minute pulverization of the water as the liquid flashes into steam, and passes off into the region of the separating chamber.

There is still an additional feature to be considered. Ordinarily, bubbles form within the surface of a liquid when the pressure of the water vapor component exceeds the surface tension of the liquid. In superheated liquids, however, there is a probability that the formation of bubbles will take place upon nuclei. Such nuclei are furnished by minute bubbles of air, which have appeared within the body of the liquid due to the reduction of solubility with temperature. When steam bubbles in the explosive process form upon these air bubbles as nuclei, they, in passing off into the region of the vapor space of the deaerator, take with them in mechanical mixture the bubbles of air. Some portion of the action of the Elliott deaerator is, therefore, a mechanical problem of vapor formation and vapor mixture.

The essential fact of course in the explosive boiling is that it is controlled, and regulated in accordance with a definite need. Ordinarily, it has been found by experience that a temperature gradient of 22 deg. between the water and the entering liquid furnishes sufficient energy for the explosive boiling and the mechanical separation. This value varies somewhat with temperature being as low as 15 deg. in cases of separator temperatures around 195 deg., but is generally not exceeded. After the explosive boiling which is relied upon to produce the bulk of the separation, the water in spray form is collected and cascaded over corrugated pans, the idea being to agitate the water in the region of reduced vapor pressure, in order to accomplish any residual separation which is possible. Operating results are secured with this method vary-

ing from zero to 1/10 cc. of oxygen per liter. It has become conventional in both chemical and engineering parlance to express oxygen or air concentration in water as cubic centimeters per liter. It is perhaps a desirable convention, for it has a definite meaning in the metric system and is convertible as parts per thousand by volume in the English system. In the explosive boiling process, heat is liberated in the separator in the form of steam; this heat being exactly proportional to the heat loss in the explosive boiling. The vapors from the interior of the separator vacuum space are conducted to a condenser which is a part of the deaerator, and there condensed, the heat being returned to the cooling water which is the supply going to the heater to be heated for admission to the separator. Dissolved gases such as air and oxygen are of course not condensable, and they are removed by some form of air pump, desirably a steam jet ejector, and delivered to the atmosphere. Since the heat involved in the vapors is returned to the cooling water, the heat loss in the process has been substantially zero.

Control is accomplished by regulation of the vacuum producing characteristics of the air pump, the desired factor being an unvarying vacuum. The condenser's primary function is condensation and heat recovery. It is first desired to return the heat involved in the boiling process to the incoming water going to the heater, in order to make the process one hundred per cent heat efficient. It is further desired to remove all vapors and return them to the separator as liquids, thus, relieving the air evacuating means of any necessity of handling vapor. The ejector or air pump, as the case may be, therefore, handles only non-condensable gases, and it controls the vacuum. The condenser, as far as it is a vacuum producing means, is inserted primarily to refrigerate the vapor mixture handled by the air pumps.

The vacuum in the condenser bears no definite relation to the mean temperature of the cooling water, but the condenser itself may take advantage of much higher temperature gradient between vacuum and cooling water than is possible in the high vacuum surface condenser. While it operates at low conductivity, it is nevertheless an efficient condensing means, on account of the non-relation between vacuum which is controlled by the air pump and cooling water temperatures which are controlled from a natural supply.

There is another feature of the condenser performances which is noteworthy. The water to be deaerated in passing through the deaerator chamber liberates a certain amount of heat, which we have specified in a previous statement as being approximately that corresponding to a 22 deg. B.t.u. drop. This represents a definite amount of vapor which goes into the condenser, and the circulation of this condenser is provided by incoming supply. The condenser, therefore, operates under a very definite circulation rate, and its duty from this standpoint is invariable, although the heat handled in gross quantities may be different, that is, in round numbers with a definite circulation rate, there is also an invariable condition of temperature, and a definite relation of steam handled to water velocity. The heat may vary, but the surface remains invariable, and if it is large enough under the maximum loading, it has a higher margin of safety under all other conditions.

The vacuum pump type, generally, as far as operation is concerned, is immaterial. Steam jet air ejectors furnish the simplest and most efficient means, since it is possible subsequently to recover all the sensible heat from the steam at its exhaust, and there is no appreciable heat energy wasted. Condensers are uniformly of the surface type, although on units handling around 1,000,000 lb. per hr., the jet condenser is employed. Ordinarily, the jet condenser is not as efficient, however, as the surface condenser, because of the necessity of a water removal pump. Either open or closed feed water heaters may be employed, and as far as deaeration is concerned, the use of either is optional. With high initial dissolved air content in water entering the deaerator, the corrosive condition of the atmosphere within the deaerator is much more severe, but it is possible to proportion the deaerator chamber to handle this corrosive condition by the use of cast iron and monel or nickel plated fittings. The open heater, however, removes a considerable proportion of the initial dissolved air and is itself a cheaper form of apparatus; hence, its use is desirable whenever possible. It has, however, particularly in locations where headroom is scarce, the disadvantage that it must be mounted above the deaerator in order to provide positive pressure on the deaerator admission valves.

In building service for the protection of domestic hot water lines, the closed heater is almost universal, for the reason that the exhaust steam supplied is almost invariably polluted by the oil from engines, and the mixture of oil with water for lavatory or cooking purposes is impractical. The closed heater also offers more leeway in the matter of location, since pressure of the supply lines may be carried through the heater up to the deaerator admission valve. The closed heater is, of course, undesirable on account of more severe corrosive conditions in the deaerator, as well as its greater cost. Vacuum control is obtained where the steam ejector is used by the regulation of steam pressure on the ejector. Where motor driven vacuum pumps are employed, the regulation is stop and start, the pump operating in accordance with the load on the deaerator. Where steam ejectors are used a small auxiliary condenser of the conventional surface type is employed to recover the heat in the ejector exhaust; not only the small proportion of vapor drawn over from the deaerator condenser, but also the operating steam heat. The vapors are vented under atmospheric pressure and the condenser operates under atmospheric pressure or with a very slight differential increase. The pump suction on the deaerator is under the same vacuum as exists in the deaerator chamber. On power plant installations, the conditions of headroom, etc., are such that ordinarily the deaerator may be elevated sufficiently above the boiler feed pumps to provide a positive head upon their suction, after the equivalent water column due to the vacuum has been deducted from the static suction head. Such pumps operate in their normal manner.

On building installations, the headroom conditions are such that there is ordinarily a vacuum upon the suction of the pump. In order to secure reliable operation, it is necessary to vent the suction of these pumps back into the deaerator vapor space and employ a pump of the hot well type. On account of vacuum on the glands, they must of necessity be pressure sealed.

Pumps may be either of the centrifugal or the duplex plunger type. The simple pump is not applicable to this service because the deaerated water immediately dissolves the cushioning air in the air chamber and produces a water hammer on the line at each stroke of the pump. If the conditions of service are satisfactory, the centrifugal pump is by far the better unit; but the duplex plunger pump, rod packed type, with water sealed rod glands is altogether a useful unit.

Simultaneously with the development of this type of apparatus it was necessary to develop methods of determining dissolved oxygen in water which would be commercially useful. Physical methods, while direct and convincing, are ordinarily so lacking in sensitivity that they are not useful when the elimination of corrosion is sought. It was accordingly necessary to turn to chemical methods of determining the dissolved oxygen in order to secure both accuracy and portability. Portable test kits in suitcase form have been developed for making tests by the Winkler method, which was adopted and modified as being both the most generally recognized in a scientific way as well as having the greatest field of usefulness. It is in general applicable to all conditions where the waters do not contain either nitrates or nitrites, and such waters, while not unusual, are very infrequent. For such waters another method known as Lenoissier's has been developed, but ordinarily is not generally employed.

The Winkler iodimetric method depends upon liberating in a solution containing dissolved oxygen an amount of iodine exactly proportional to the original content of oxygen. The iodine may be determined by a volumetric standardized chemical solution with a starch indicator, and is accurate in numerical figures to 12 parts in 100 million by weight. The deaerator unit operating below 180 deg. falls below this limit, the basis of the guarantee being either zero or a trace. The minimum value in cc. per liter which it is possible to determine accurately is 0.01 cc., and anything less than this is designated either as zero or a trace, the zero value being so low that the chemical indicator which is starch solution is not able to detect it, and the iodine starch indicator is perhaps one of the most sensitive in chemistry.

It is unusual, perhaps, in engineering practice to deal with such small quantities, yet corrosion is a continuous accumulative process and the minimum results which are obtained are worth-while over the period of expectation of life from standard engineering equipment.

DISCUSSION

T. M. DUGAN: I can affirm many of the statements made by Mr. McDermet in regard to the life of various kinds of pipe. I, personally, have had a number of experiences along that line particularly among failures of pipes in domestic use.

As engineers we should keep ourselves in touch with the many apparatuses in use for the protection of pipe. The results obtained from their use are absolutely remarkable.

I thought perhaps a sample of evidence might carry it home to you a little stronger so I asked to have a sample of pipe sent to me but unfor-

tunately it has not been delivered. It was a piece of black pipe from a plant installed in 1915 in a public building in the city of Pittsburgh where we have a scrap iron deactivating apparatus. The pipe has been in use for three years and shows practically no dissolution. The average removal of the dissolved oxygen in this case was about 90 per cent.

Mr. McDermet illustrated some of the ideas very distinctly and I think covered my point as well as if I had the pipe.

During the past 12 years I have made many practical and experimental tests for the removal of dissolved oxygen from water and can assure you the methods explained by Mr. McDermet are very efficient and commendable.

I hope to see in the near future an apparatus that can be used in the smaller type buildings that do not require an engineer to control them. For you know many of our apartment houses, dwellings, etc., are having frequent trouble from red rust water and corroded pipes.

During the month of January while making some observations of deactivating systems in use in Philadelphia, I visited one at the Gerard Estate, which many of you also visited during the convention held in Philadelphia a year or more ago. This is a very large plant and protects almost seven miles of hot water piping much of which is under ground. It has been in operation about two years. The engineer in charge told me that before its installation they had as many as fifty repairs to the piping in a month and now only four or five. In addition to this the water has become clear and the pipes cleaned out to such an extent that they were able to dispense with the use of an extra pump which had to be used on account of the pipes being clogged.

Mr. McDermet's illustration of clogged pipes gives a wonderful idea of how the interior of their pipe looked before the installation of the deactivator. The engineer further told me the saving to their equipment and operation of same per year was many thousands of dollars.

Again I say we should give the subject, Corrosion of Pipe, very serious thought.

F. N. Speller (written): This paper again brings up in a forceful way the old trouble with keeping hot water pipes in service. Many have gotten so used to this trouble that it is taken as a matter of course and as a legitimate field for the plumber to exercise his art It is surprising how much of the brass pipe used in hot water lines fails in 10 or 15 years of service. This is due to dezincification and is not manifested by clogging of the lines or discoloration of the water. The first sign of trouble is either a whitish incrustation on the outside of the pipe at places where the water oozes through or by breaking off at the threads due to weakening of the interior wall of the pipe—often there is little more than spongy copper under the thread. This, as I have shown in a previous paper, may be greatly retarded by removing a considerable portion of the free oxygen, but this must be taken in time or the pipe will not be worth saving. Corrosion affects only the surface of iron pipe but disintegrates the brass so that it is left, after a while, in a greatly weakened condition with little ductility. This is a very important thing to keep in mind. Those in charge of buildings should arrange a by-pass on

the hot water lines near the heater so that the deterioration of the pipe can be determined and remedial measures applied before it is too late.

It has been proven beyond a doubt that dissolved oxygen is responsible for corrosion and that corrosion is approximately proportionate to the free oxygen in the water, temperature and other factors being the same. Any practical form of apparatus which will reduce the free oxygen to about 0.4 c.c. per liter will serve to protect hot water service lines.

Apparatus for removing oxygen may be divided into two classes: (1) those in which the water is heated under low pressure so that all the gases are discharged; (2) those in which the hot water is kept in contact with a large surface of iron under high pressure by which only the free oxygen is removed and fixed as hydroxide of iron, commonly known as rust.

The appliances that have been used so far have been designed more particularly for large buildings. Recent developments, however, indicate that protection will soon be applied economically to smaller buildings including the average residence.

No. 630

BODY TEMPERATURES AND THEIR MEASUREMENTS

By W. J. McConnell[1], Pittsburgh, Pa. (Non-Member)
And
F. C. Houghten, Pittsburgh, Pa. (Member)

As part of an extended and painstaking investigation into the physiological effects of high temperatures and humidities now being conducted by the United States Public Health Service and the Bureau of Mines, cooperating with the American Society of Heating and Ventilating Engineers, over 1,000 observations of body temperatures have been made to date.

While no definite conclusions have been drawn from these observations, a brief review of the subject of body temperature, together with a consideration of findings and methods of measuring body temperatures may serve at this time to invite a discussion among those interested in this field of research.

A knowledge of the normal range of body temperatures, and fluctuations from the normal allowable from an economic and humanitarian point of view in persons working almost continuously under conditions conducive to a rise in temperature, is of immeasurable value as an index of the physical conditions of the body.

Many previous critical scientific investigations along these lines are misleading because of the inaccurate methods heretofore available for measuring the temperature of different portions of the body. Frequently only the mouth temperature was considered. With the advent of new instruments of precision for measuring temperatures of the body this source of error has been largely overcome, and a section of this paper is devoted to a description of accurate methods used in the present investigation.

The subject of body heat, itself, has provoked much disputation among early investigators. J. Hunter, the celebrated surgeon and anatomist, taught that the mysterious principle of animal heat resided in the stomach. Barthez and his followers attributed heat production to the commingling of the several liquid and solid portions of the organism. Haller, in 1757 believed that the friction of the circulating blood and movements of the heart and blood vessels caused the heat of the body. Innumerable experiments followed these teachings. Beginning with the classic experiments

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Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

of Lavoisier, the French chemist, in 1775, and those of Hermanns in 1883, it has been shown to the satisfaction of all that most of the energy spent in the body, of which muscular work (either voluntary or involuntary) is the important source, finally becomes heat. The energy is derived from the food, drink, and air consumed and transformed by chemical processes of a vital nature into available energy for use in the body. The amount of heat produced varies according to the character and extent of the chemical processes within the tissues.

Heat is lost from the human body by radiation, conduction, convection, and evaporation from the body, and to some extent is carried away in the body exertions. Just how body temperature is controlled is not yet clear. It has long been assumed that in the body there exists a special set of heat nerves and heat centers, but no evidence of a positive nature of such centers has been produced. Perhaps the more conservative view to adopt is that expressed by Pembrey and others, that heat regulation is not controlled by any special heat regulating apparatus composed of heat centers and heat nerves, but is associated with coordinated activity of the nervous system controlling the muscles, glands and blood vessels in response to changes of temperature.

The principles involved in the production and elimination of heat in the human organism, therefore, compare in many ways with mechanical means devised for heating purposes. The fuel of the human organism is derived from the food, water and air consumed. By the process of oxidation (or slow combustion) which takes place in the body tissues, these are transformed into energy, which finally becomes heat. This heat in turn is given off to the environment largely by the physical methods above named. A certain amount of heat produced in a mechanical device is lost in chimney gases and ashes and in like manner a small proportion of heat is lost in the human subject, through expired air and body excretions.

A certain rhythm in body temperature has been recognized by physiologists for a long time and according to Benedict and Slack the minimum temperature occurs during the early hours, usually between 2 and 3 o'clock, followed by a marked early morning rise which becomes less pronounced as the day progresses, and again reaches its maximum in the afternoon about 5 o'clock. This is followed by a slight fall which becomes very noticeable after retiring and gradually continues until the minimum point is again reached in the early morning.

No doubt many factors modify the temperature rhythm of the body and the experiments of Galbraith and Simpson indicate that the temperature rhythm can be reversed. They showed that in monkeys the rhythm may be completely inverted by inverting the monkey's routine of life. Benedict did not find any inversion of body temperature in men who slept during the day and worked at night though the temperature curve was modified (see Gibson, *Amer. J. Phys.*, 1905); sleep and rest during the day lowered the body temperature, but work at night did not appreciably raise it. Linehard, however, found a reversal of rhythm in the Arctic region when he changed the hour of lighting the lamps, reversing day and nights of the polar night. Gibson also found this reversal of temperature rhythm in himself during his travels around the world.

Some comparative studies of rectal temperatures with mouth tempera-

ture have been made showing how unreliable mouth readings are. Very conclusive evidence in favor of rectal readings is given by Benedict and Slack.

The abnormal conditions produced in the body by combined high temperature and humidities have furnished an interesting problem for research. In 1905 Haldane and his collaborators began their carefully planned studies of this subject. In America the New York Commission of Ventilation has for some time been engaged in a similar study.

The aims of our investigation of the problem were given in detail by Dean Anderson, at a recent meeting. Notwithstanding many difficulties which had to be overcome and time lost in remediating defects detected in the pychrometic chamber, work has progressed.

Before beginning experiments under the controlled conditions of temperature, humidity, and air motion, a number of observations were made on individuals who later were to take part in the experiments. Especially it was desirable to learn, if possible the temperature rhythm of these men and to compare temperature readings measured in the mouth with those in the rectum.

In this preliminary study, the mouth temperature was measured by means of "one minute" clinical mercury thermometers and the rectal temperature was measured by the short bulbed "one minute" thermometer of the same make. In some cases readings were made at the same hour every day and in others at different hours, usually between 9 o'clock in the morning and 4 o'clock in the afternoon.

A study of these measurements, while indicating a general tendency for the temperature to rise in the afternoon, show no degree of constancy from day to day in either the temperature taken by mouth or rectum.

Frequently a higher temperature was found in the morning than in the afternoon. Temperature reading taken at the same hour every day likewise varied considerably.

While accurate measurements of the temperature and humidity of the room, where body temperatures were taken, were not kept, and in spite of what might be expected through the influence of temperature rhythm, it was observed that the low afternoon temperatures corresponded to a noticeable degree to the temperatures of the room which was cooled by opening the windows, or else occurred after the subject entered from the outside on a cold day, and the high morning temperature occurred after resting in a warm room after a brisk walk.

The lowest temperature registered by rectum was 98.2 deg. fahr., mouth 97.1 deg. fahr., and the highest 99.8 deg. fahr. and 100 deg. fahr., respectively.

This shows a fluctuation of temperature in the rectal of 1.6 deg. fahr. while in the oral there is a fluctuation of 2.9 deg. fahr.

Mouth temperatures are extremely irregular and misleading and should be discarded wherever accurate work is desired. In experiments which followed it was found that in hot and humid air the two curves tend to approximate and occasionally the mouth and rectal temperatures became identical. When the many factors which may conceivably change body temperature are considered, the parallelism between the temperature and

humidity of environing atmosphere and the body temperatures is very remarkable.

So far the experiments have been confined to room conditions ranging from 75 deg. fahr. dry, 63.5 deg. wet, relative humidity 40 per cent to 100 deg. fahr. dry, 98 deg. wet with 93 per cent relative humidity in nearly still air, and observations were made on subjects at rest.

It is only possible at this time to give the general trend of the findings and a more detailed account with tables of readings recorded is reserved for a later paper.

After a few experiments it was determined that other factors besides temperature rhythm exerted an influence on the body temperature. For example, in one experiment where the dry bulb registered 90 deg. fahr. and the wet 87 deg. fahr. one subject's rectal temperature was 99.7 (mouth 99.0) at 1.53 p. m., beginning of the experiment, and 100.2 deg. fahr. rectal (mouth 99.2) at 4.17 p. m., the end of the experiment; another subject's was 100.2 (mouth 100) at the beginning and 99.8 (mouth 99.4) at the end, while a third subject's rectal temperature was 100 deg. (mouth 100) at the beginning and 101.0 rectal (mouth 100.2 deg.) at the end.

The temperature of the subject that rose during that experiment dropped from 100 to 99.8 deg. fahr. rectal temperature (mouth 98.9 to 98.6) in another experiment when the dry bulb was 86.3 deg. fahr. and wet 85 deg., while the subjects, whose temperature dropped in the first experiment remained constant in the second. This occurred in a number of subsequent experiments although the temperature and humidity were higher in the chamber than outside. Changes in metabolism and pulse rate also occurred. These experiments indicate that the body temperature may drop in an atmosphere of a higher temperature and humidity during a period of rest.

To obviate this difficulty, it was found advisable to require the subjects to rest for a period of two hours in a room of ordinary temperature and humidity before entering the controlled chamber. In the few experiments so far completed under these conditions the body temperature dropped during the period of rest and on entering the controlled chamber the body temperatures usually began to rise whenever there was a temperature and humidity in the chamber above 85 deg. fahr. wet bulb.

The subjective and objective effects produced in the body by the combined elevations of temperature and humidity of the air, seem to depend to some extent on other conditions than the rise of body temperature per se, and investigation into other influencing factors is necessary before the harmful effects of marked fluctuations of body temperature can be evaluated adequately.

MEASUREMENTS OF TEMPERATURE BY VARIOUS METHODS

In order that one may judge of the accuracy of the thermo-electric method which is used for measuring body temperature, the principles involved in the methods and their application are here discussed.

The thermo-electric method of measuring temperatures depends upon the fact that whenever two metals are in contact an e.m.f. is set up, the

magnitude of which depends upon the temperature of the point of contact. A measure of this e.m.f. is therefore a measure of the temperature of the junction of the two metals. A thermo-couple is made by joining one end of each of two wires of different metal or alloys, as copper and constantan. These ends are usually joined by welding. The junction of these wires is placed at the point whose temperature is to be

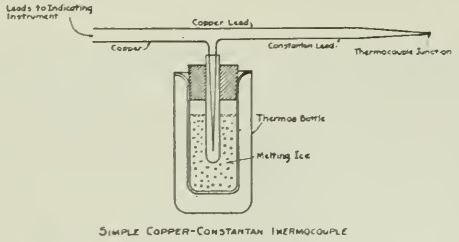

FIG. 1. SIMPLE COPPER-CONSTANTAN THERMO-COUPLE

measured. Fig. 1 shows such a couple. The end of the copper wire goes directly to an instrument which indicates e.m.f. preferably a potentiometer. The other end of the constantan wire is joined in a constant temperature bath, preferably an ice bath, to a second copper wire, which also goes to the indicating instrument. The e.m.f. indicated depends on the difference in temperature between the two junctions; and with a known constant temperature, the junction can be calibrated to indicate temperature.

For measuring rectal temperatures the wires at and near the junction were surrounded with plastic rubber placed in a mold to give the indicated shape, Fig. 2, and then vulcanized to the desired hardness. The

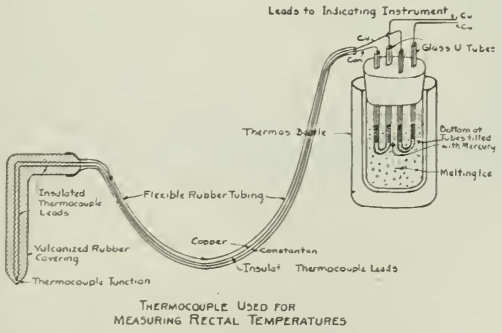

FIG. 2. THERMO-COUPLE FOR RECORDING RECTAL TEMPERATURES

insulated leads from this vulcanized part extending about 3 ft. are enclosed in a flexible rubber tube. This flexible tube joins onto the vulcanized part. When it is desired to take a temperature, the subject makes electric contact between the leads and the copper wires leading to the potentiometer by inserting the leads in glass tubes containing a small amount of mercury. The mercury makes electric contact between the

leads and the wires leading to the potentiometer. These glass tubes are surrounded by ice in a thermos bottle. A Leeds & Northrup precision potentiometer located in another room, Fig. 3, is used to indicate the e.m.f.

These thermo-couples were calibrated by placing them together with a standard mercurial thermometer and also a clinical thermometer which had previously been used for these measurements in a mercury bath suspended

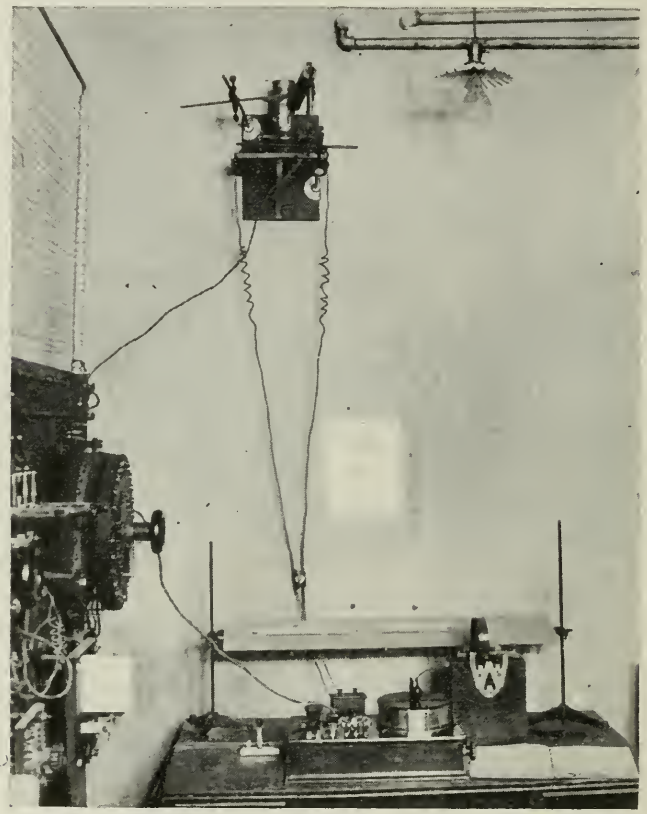

FIG. 3. PRECISION POTENTIOMETER USED TO INDICATE E.M.F. IN MEASURING BODY TEMPERATURE WORK

in a thermos food jar filled with water at body temperature. A large number of observations showed a marked consistency between the thermo-couple and the standard, but a variation of as much as 0.2 deg. between the clinical thermometer and the standard.

The measurement of skin temperatures is attended with all the difficulties usually encountered in determining surface temperatures. It is practically impossible to determine the true surface temperature of any body for the reason that the active part of any pyrometer which may be used, cannot be placed in the geometrical surface. Approximations may be made which may give their temperatures within very close limits. An

ideal pyrometer to be used for the measurement of surface temperatures would be one with an infinitely small active element.

A thermo-couple made of a very fine wire most nearly approaches this ideal and is therefore the only logical instrument for this use. In determining surface temperatures the thermo-couple is often placed under the surface. This is obviously impossible in determining the temperature of human skin, and it is therefore necessary to attach the couple to the surface of the skin. The surface of the human body is heated from within by means of a circulating fluid, the blood, which is at an average temperature of 98.6 deg. fahr. The surface loses heat by radiation, convection, and evaporation. Anything which interferes with the normal loss of heat by one of these means will, to that extent, help create an

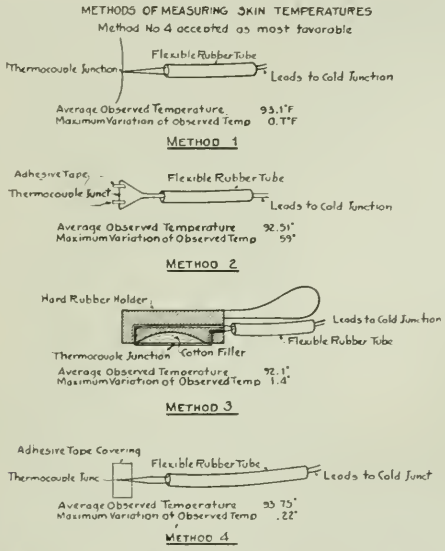

FIG. 4. METHODS OF MEASURING SKIN TEMPERATURES

error in the observed temperature. On the other hand any instrument placed on the surface and touching it on only one side while exposing any considerable area to the air at lower temperature will not indicate the true skin temperature.

The object in determining skin temperature in this investigation has been:

1. To determine as nearly as possible the actual temperature of the skin.
2. To indicate body reactions to the immediate environment of the subject.

In these investigations as has been stated the latter has been considered the more important. It was therefore more desirable to accept a method of determining those temperatures which would give the most consistent readings even though those readings were slightly above or below the true temperatures rather than a method which would give readings whose average was more nearly the true surface temperature, but which shows a meaningless variation between consecutive readings.

Four methods of applying the thermo-couple to the body have been investigated.

1. The thermo-couple junction was applied as in Fig. 4, Method 1 and held in place until the indicated temperature showed no change. By this method indicated temperatures varying by 1.9 deg. were obtained by varying the pressure of contact, as indicated in Table 1. By holding the couple in place with as nearly the same pressure as possible, indicated temperatures varying by 0.7 deg. were obtained.

2. The junction was applied to the surface as in Fig. 4, Method 2. This method indicated temperatures varying by 0.6 deg., when the thermo-couple was placed as nearly as possible, in the same identical manner or after the subject has moved the parts of the skin on which the couple was placed. Table 2 gives a series of such observations and the manipulation of the thermo-couple between readings.

FIG. 5. THERMO-COUPLE IN PLACE ON CHEEK AND BACK

FIG. 6. THERMO-COUPLE ON CHEEK AND ABDOMEN; ALSO LEADS FROM RECTAL THERMO-COUPLE

3. The thermo-couple was attached to a vulcanite block but insulated from it by means of cotton. This block was applied to the skin so that the thermo-couple junction was pressed between the surface of the skin and the cotton as in Fig. 4, Method 3. With this method the indicated temperature rose rapidly for about 20 seconds after applying the couple to the skin. Consecutive observations vary by 1.4 deg. as shown in Table 3.

4. The thermo-couple junction was fastened to the skin by means of a piece of adhesive plaster about ½ in. square and left in that position throughout the test. Fig. 4, Method 4. Observed temperatures taken by this method after removing and replacing the thermo-couple, under the same conditions, vary by 0.3 deg. after rubbing the hand over the plaster and moving the skin, the indicated temperatures varied by 0.2 deg. as shown in Table 4. All observed temperatures in Tables 1, 2, 3 and 4, were taken under the same atmospheric conditions and at nearly the same location on the skin of the same subject. The actual skin temperature, we may therefore assume, was nearly constant, and the variations in the observed temperatures are due to errors in the various methods.

TABLE 1—METHOD 1

Temperature of Skin of Cheek as Indicated by Thermo-couple Applied as in Fig. 4, Method 1

Time	Temperature	Average Temperature	Variation from Average
10:42	92.0		
10:43	92.5		
10:44	92.8		
10:45	93.0		
10:46	93.5		
10:47	93.6		
10:48	93.7		
10:49	93.9	93.1	
10:50	93.0		0.1
10:52	92.7		0.4
10:56	93.0		0.1
10:58	93.4		0.3
11:00	93.8		0.7
11:02	92.5		0.6
11:04	93.0		0.1
11:06	93.1		0.0
11:08	92.9		0.2
11:10	93.4	93.1	0.3

Temperatures taken between 10:42 and 10:49 by applying the couple with increasing pressure.

Temperatures taken between 10:49 and 11:10 by applying the couple with the same pressure each time, as nearly as possible.

TABLE 2—METHOD 2

Temperature of Skin of Cheek as Indicated by Thermo-couple Applied as in Figure 4, Method 2

Time	Temperature	Average Temperature	Variation from Average
11:51	92.04*		0.47
11:52	92.19*		0.32
11:54	92.77*		0.26
11:55	92.41†		0.10
11:56	92.41†		0.10
11.58	92.41†		0.10
12:00	92.31†		0.20
12:02	93.01†		0.50
12:03	93.10†	92.51	0.59

* Couple removed for instant before making observation.

† Cheek muscle moved before making observation.

TABLE 3—METHOD 3

Temperature of Skin of Cheek as Indicated by Thermo-couple Applied as in Fig. 4, Method 3

Time	Temperature	Average Temperature	Variation from Average
11:20	90.7		1.4
11:23	92.1		0.0
11:26	91.2		0.9
11:29	93.4		1.3
11:32	92.8		0.7
11:35	92.3		0.2
11:39	91.3		0.8
11:42	92.8	92.1	0.7

TABLE 4—METHOD 4

Temperature of Skin of Cheek as Indicated by Thermo-couple Applied as in Fig. 4, Method 4

Time	Temperature	Average Temperature	Variation from Average
10:06	93.73*		0.02
10:08	93.46*		0.29
10:10	93.74*		0.01
10:12	93.85*		0.10
10:14	93.64*		
10:16	93.96*		0.21
10:19	93.70*		0.05
10:21	93.54*		0.21
10:23	93.96*		0.21
10:25	93.92*	93.75	0.17
10:26	93.70†		0.05
10:27	93.75†		0.00
10:28	93.71†		0.04
10:29	93.62†		0.13
10:30	93.70†		0.05
10:31	93.67†		0.08
10:33	93.56†		0.19
10:34	93.70†		0.05
10:35	93.91†		0.16
11:15	93.80†		0.05
11:17	93.73†		0.02
11:45	93.84†		0.09
11:46	93.80†		0.05
11:47	93.70†		0.05
12:09	93.97†		0.22
12:10	93.82†	93.75	0.07

* Couple removed for instant before making observation.

† Cheek muscles moved and thermo-couple pressed down slightly by finger before making observation.

It is evident from the tables that the fourth method gives the greatest consistency between individual observations. This method also gives the highest indicated temperatures, as would be expected, for the reason that the plaster interferes with the loss of heat from the skin. The second method gives a lower indicated skin temperature which is no doubt below the true skin temperature for the reason that a large part of the couple is exposed to the air. The other two methods give indicated temperatures which vary a great deal. The fourth method was therefore accepted as giving the most consistent results. Temperatures observed by this method must be within a fraction of a degree higher than the true skin temperatures, since they are only a degree higher than temperatures observed by the second method which are certainly lower than the true skin temperatures.

No. 631

STANDARD METHOD OF TESTING DUST REMOVAL EFFICIENCIES OF AIR WASHERS

By O. W. Armspach and Margaret Ingels, Pittsburgh, Pa.
Members

WITH A NEW METHOD OF MAKING AIR DUST DETERMINATIONS

By F. Paul Anderson and O. W. Armspach, Pittsburgh, Pa.
Members

This investigation will be reported in three sections, of which the first part is here presented. Part I considers Methods of Testing, Part II will take up Relative Efficiencies of Washers With Various Dusts, and Part III will present a Standard Test Code.

PART I

Methods of Testing

Introduction

In developing a standard method of determining the air cleansing efficiency of air washers the writers found very little published data on the subject aside from the reports of the American Society of Heating and Ventilating Engineers Committee on Standard Method of Testing Air Washers (see Transactions, A. S. H. & V. E., 1914, p. 425 and 1917, p. 407). The Committee encountered and overcame many difficulties in their investigations which considered practically all known methods of determining the dust content of air. These methods when applied to washer testing involve additional difficulties inasmuch as two samples must be taken simultaneously and the samples must represent average conditions over the cross section of the washer. The Committee concluded that most of the known methods would not meet these requirements, or they were in other respects unreliable and inaccurate. Therefore, several methods can be eliminated from the investigation as being impractical and lacking the accuracy which is believed desirable and possible.

It is the object of the investigation to develop a definite and complete standard code for testing air washers under different conditions of air

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

dustiness. Tests show that the efficiency will vary widely, depending upon the nature of dust which enters the washer. Hence, it is proposed to develop an AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS standard dust so that tests when desirable, can be on an equal and representative basis. This composite dust will be made up of definite percentages of various kinds of particles. Washers installed in the industries are often required to handle a definite dust such as wood, glass, sand, street dusts, or various metallic dusts. It is therefore desirable also to test washers when handling these specific materials so that relative efficiencies will be available for specific installations.

DUST DETERMINATION METHODS

The Sedgwick-Rafter Counting Cell: This cell is at the present time being used to considerable extent to make quantitative determinations of air dust from samples collected under water or with the sugar tube filter. In view of its extensive use it was thought that the method could possibly be applied to tests of air washers, and a description of the method and results obtained is included in this paper.

The cell consists of a 20 × 50 mm. metal frame 1 mm. deep mounted on a 1 × 3 in. glass slide. It holds 1 cc. of solution. A dust sample is obtained by passing a known quantity of air through a sugar or other soluble filter. The sugar is then dissolved in 500 cc. of distilled water and, to facilitate counting, 2 cc. of this solution is added to 20 cc. of distilled water.

One cubic centimeter is then transferred to the cell with a clean pipette, and a thin glass slide is placed over the same to protect the sample from further contamination. The whole is placed under the microscope for counting. The microscope is adjusted so that the field will cover 1 sq. mm. and the number of particles in one-fourth of this field is determined. This number is multiplied by 4,000 to obtain the number of particles in 1 cc. of the solution. This again multiplied by 22 and divided by 2, the dilution factor, gives the number of dust particles contained in 1 cc. of the original solution. Since 500 cc. of water was used to dissolve the sugar, these figures are further multiplied by 500 to obtain the total number of dust particles caught by the filter. The result is then divided by the cubic feet of air passed through the filter and the result gives the number of dust particles contained in a cubic foot of air.

The formula expressing this relation is as follows:

$$D = \frac{d \times 4000 \times 500 \times 22}{V \times 2}$$

Therefore the dust particles per cubic foot equals:

$$D = \frac{d \times 22{,}000{,}000}{V}$$

where

D = number of dust particles contained in one cubic foot of air
d = particles counted in one-fourth field of the microscope
V = cubic feet of air used to obtain the sample

From the foregoing it will be observed that the multiplying factor is 22,000,000 and any error in counting is multiplied to just that extent. Furthermore, with a multiplying factor of 22,000,000 an assumption is made at the outset that the air to be tested contains at least this number of particles, since it is impossible to count a fraction of a particle. For air of a high dust count it is sometimes necessary to use even a greater dilution factor and the multiplying factor may be much larger.

In order to determine the effect of this multiplying factor a sample of air was obtained with the sugar tube, and then several counts using different dilution factors were made on this same air sample. Correctly, the different counts should give exactly the same result inasmuch as they were made on the same sample. Only the dilution factor differed. The results of the count are shown in Fig. 1 and proved very illuminating. The

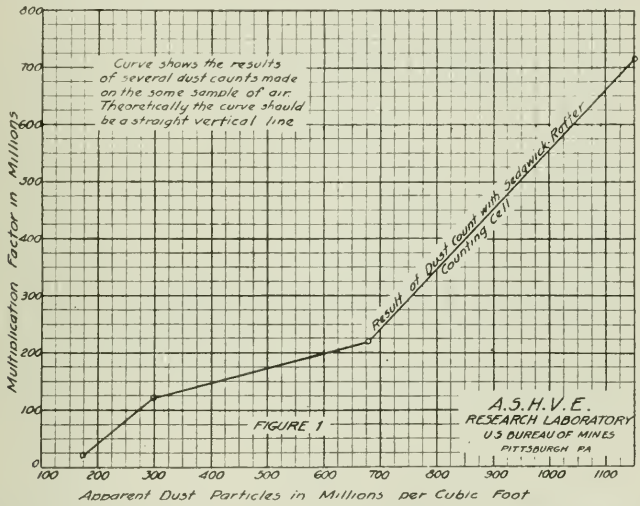

FIG. 1. RESULTS OF SEVERAL DUST COUNTS ON THE SAME SAMPLE OF AIR

curve shows that the multiplying factor, to a great extent, determines the apparent dust content, and this dust count increases with increased multiplying factor. The experiment demonstrates that the counting cell is very unsatisfactory for making air washer tests. The method was therefore abandoned and a new and satisfactory method has been developed.

A thorough examination of existing methods of dust counting has shown that the results which can be obtained vary for the same air sample, often to several hundred per cent, and are either unreliable or can be affected by the method of manipulation. In addition to this they require considerable skill and experience on the part of the operator. It was therefore decided to try to develop a new method which would conform more to the skill of probable users and which would give a direct reading for the whole sample instead of necessitating measuring the dust in an extremely small fraction of the volume of air drawn off.

The method described in the following was developed at the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS laboratory and ful-

fills the desirable conditions outlined above. It has not yet been in use long enough to fully analyze the possible percentage of variation or error, but the work which has been done shows that it gives consistency in results far higher than methods previously suggested, and involve, only manipulation and apparatus such as is commonly used by engineers.

Standard Filters: Fig. 2 shows the standard cloth filter which is now being used for making dust determinations. The filter consists of two pieces of 5/8 in. brass tube each having a 3 in. square flanged end. A standard piece of fillter cloth is placed between the two flanges and made

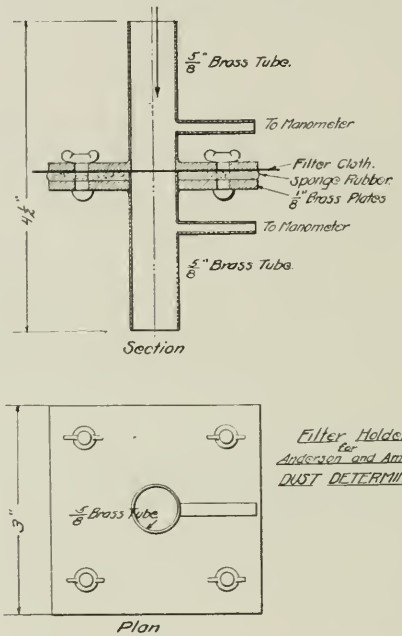

FIG. 2. FILTER AND HOLDER FOR ANDERSON AND ARMSPACH DUST DETERMINATOR

air tight by a piece of 1/4 in. sponge rubber and the whole is firmly held together by four wing nuts. A 1/4 in. tube is located on either side of filter and communicates with the 5/8 in. tubes. Rubber tubing is taken from these 1/4 in. connections to a suitable manometer which records the resistance through the filter.

The air to be tested is drawn through the instrument by means of a small air suction pump. The manometer indicates the initial resistance of filter. As the filter becomes clogged with dust the resistance shown by the manometer increases and the rate of increase is a direct and visible measure of the air dustiness. The rate at which the filter resistance changes is quite rapid even in air that is apparently quite free from dust. With these instruments it is possible to measure variations in air dustiness over 10 minute periods, and consistent variation in dust content between morning and afternoon tests have been obtained in the labora-

tory where the air is comparatively free from dust. The chief value of this determinator is that the readings obtained are exceptionally consistent and entirely independent of the personal element.

The sensitiveness of the instrument was demonstrated by an unplanned experiment. In an adjoining room a small quantity of gun powder was ignited in order to generate a smoke necessary for studying air currents. At the same time tests with the dust filters were being made and a slight but regular increase in filter resistance was being recorded. Suddenly the resistance increased surprisingly. This effect could not be accounted for, however, until an odor of burned gun powder was sensed. Minute particles of suspended powder floated from the adjoining room to the

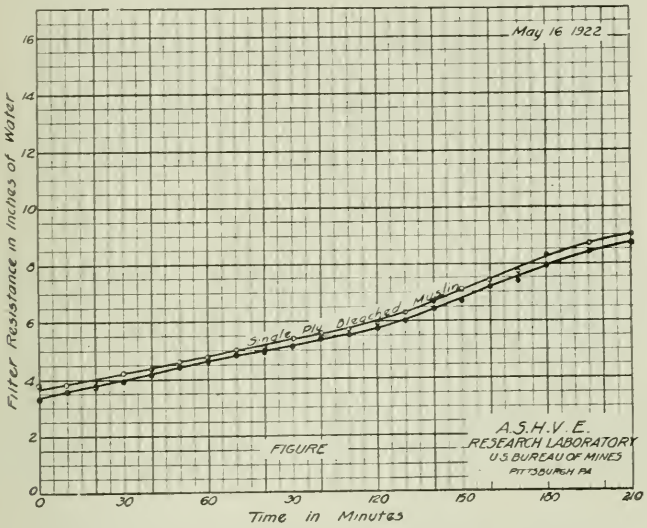

FIG. 3. RESULTS OF SIMULTANEOUS TESTS, USING SINGLE-PLY FILTER

dust apparatus and were readily detected by the determinator. The resistance increased 9 in. of water in 16 minutes.

RESULTS OBTAINED WITH STANDARD FILTER

All of the tests reported in this paper were made by passing 1 cu. ft. of air per minute through a 5/8 in. diameter bleached muslin filter. No attempt will be made at this time to give complete specifications for the cloth since it is desired to test other filters in order to find the one best suited to the requirements.

Under the microscope the filter cloth samples show that the filter will hold the relatively small particles as well as the large particles. The resistance through the filter is increased by the dust which clings to the extreme edges of the individual threads of which the filter is composed. The particles gradually fill the interstices in the cloth and the passage of air is obstructed. Although the size of the opening between the threads is many times larger than a particle of dust, still they are caught by minute hairs on the fibres, static charges or possibly other causes.

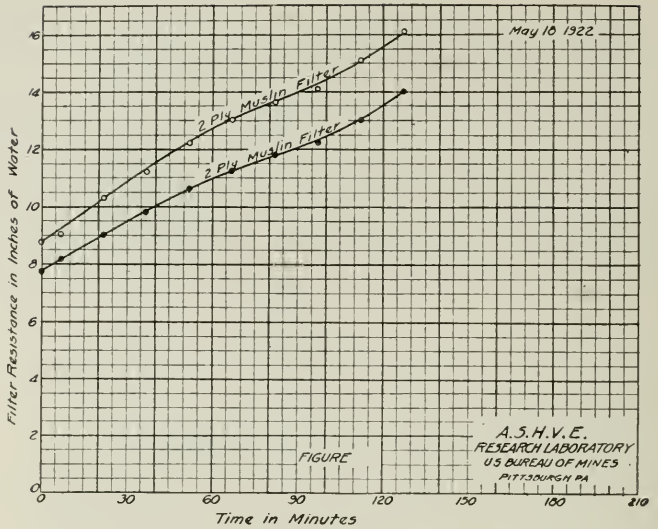

FIG. 4. RESULTS OF TWO TESTS, USING 2-PLY FILTER

Fig. 3 shows the results of two tests made simultaneously on May 16 using a single ply filter. Note that although the initial resistance varies slightly, the rate of change of resistance is the same for both filters. Fig. 4 is a test made on May 18 using a 2-ply filter. The slope of these curves indicates that there was more dust in the air on that day than on May 16. Fig. 5 is the result of simultaneous tests with a single ply filter and a 2-ply filter. The initial resistance of course is less for the single ply. However, the curves show a similarity in rate of change over a

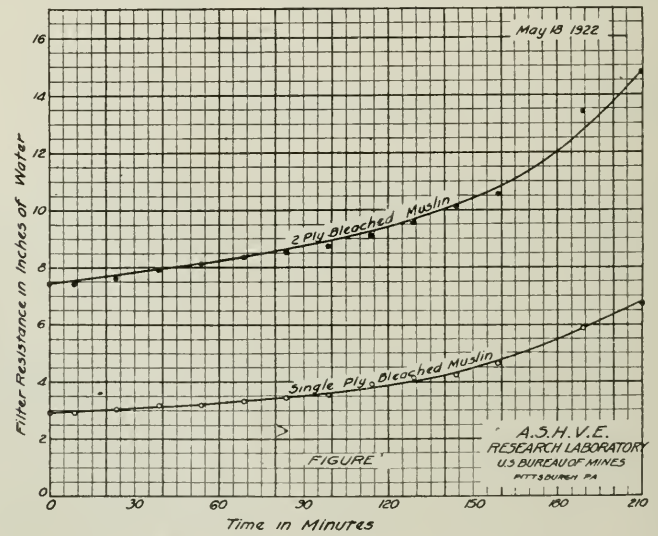

FIG. 5. RESULTS OF SIMULTANEOUS TESTS, USING SINGLE AND 2-PLY FILTERS

period of 3½ hours. Fig. 6 shows the variations in dustiness of relatively clean laboratory air. The rapidly increasing resistance compared with other tests indicates air with a higher dust content.

All the foregoing tests and many others made in a room where dust was not artificially liberated show that the filter can be adapted to dust deter-

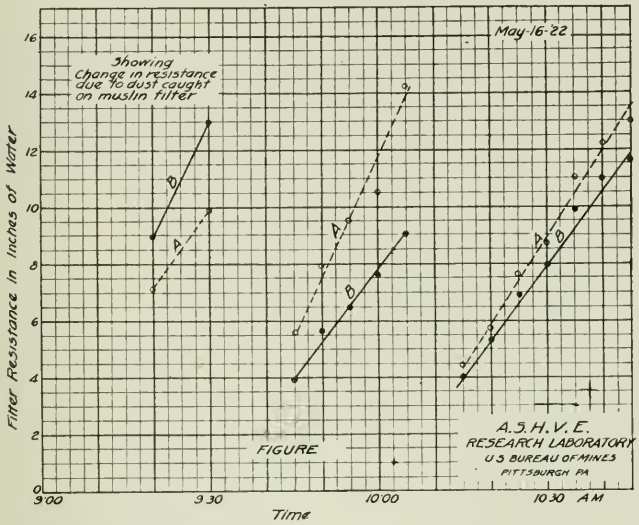

FIG. 6. CHANGE IN RESISTANCE DUE TO DUST CAUGHT ON MUSLIN FILTER

minations with considerable accuracy. The method can be used in the study of dusty industries with a considerable saving of time since the change in resistance would be quite rapid and no further procedure is necessary in the laboratory. The chief advantage of the method is that the personal equation is entirely eliminated and all that is required to obtain the relative dustiness is to read the height of a water column.

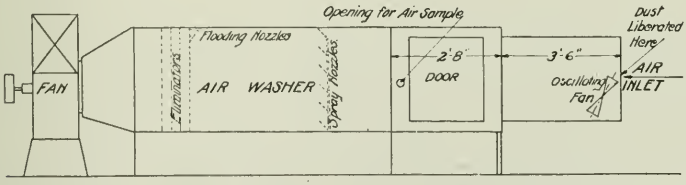

FIG. 7. EXPERIMENTAL APPARATUS FOR AIR WASHER TESTS SET

STANDARD FILTERS APPLIED TO AIR WASHER TESTS

In applying the standard filter to air washer tests it is necessary to make arrangements for taking samples of air from two points. The first sample will show the quantity of dust in air entering the washer and the second the quantity of dust in the air after it has been washed. From these two samples it is possible to calculate the efficiency of the washer as a dust

remover. The samples are taken simultaneously and at the same elevation in the apparatus.

The experimental apparatus necessary for developing an air washer code is shown in Fig. 7. A standard washer is connected to a fan handling 3100 cu. ft. of air per min., belt driven by a 2 h.p., 440 volts, 3 phase, 60 cycle motor. To the inlet of the washer is attached an ample length of duct so that the various dust materials may be liberated at a uniform rate and equally distributed over the cross section of the washer. Air passes in a straight line through the entire apparatus up to the point at which it enters the fan. A duct 6 feet long is connected to the discharge side of the fan. Water is supplied to the spray nozzles by a recirculating pump, belt driven by a 2 h.p. motor. Provision is made to raise the temperature of the spray water with a steam heater, or the temperature

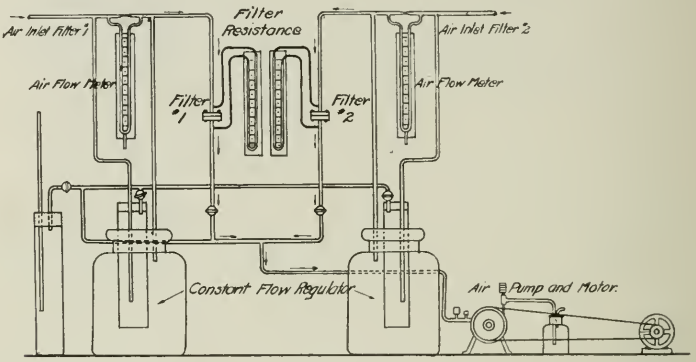

FIG. 8. DIAGRAM OF AIR SAMPLING APPARATUS USED

may be lowered by means of cooling coils from a 4-ton ammonia compressor.

Fig. 8 is a diagram of the air sampling apparatus. Air passing through test filter No. 1 is taken from the inlet end of the washer and sample No. 2 is obtained from the outlet end. A manometer is connected across each filter to show the change in resistance caused by obstruction of the dust. The quantity of air passing through each filter is indicated by a flow meter consisting of a small nozzle across which a water manometer is connected. This nozzle was carefully calibrated by means of a gas meter so that a pressure drop indicated on the manometer is equivalent to a definite volume of air. For these experiments the nozzle is calibrated to handle 1 cu. ft. of air per min., and so long as the pressure indicated by the flow meter manometer remains constant 1 cu. ft. of air will pass through the filter each minute.

The dust particles caught by the filter will increase the resistance to air flow and therefore less than one cubic foot per minute would pass through the filter. To compensate for this effect it was necessary to provide a constant flow regulating device shown in Fig. 9. The suction of the pump necessary to draw air from the washer acts at a point *C*. Jar *J* is a wide-mouth 5-gal. bottle nearly filled with water. A glass tube *G* 1½ in. in diameter passes through a rubber stopper to a point

about 1 in. above the bottom of the bottle. Another glass tube H passes from G to the inlet side of the flow meter. Tube L passes from J to the outlet side of the flow meter. Line E from the pump suction is connected to air spaces in G and K. At Point F is a tube extending below the water level in vessel K. The depth of this tube fixes the suction pressure and keeps it constant. The operation of the compensating regulator is as follows:

The valve on the suction side of the air pump is adjusted (with a maximum filter resistance) until a bubbling of air occurs from H into G. Bubbling will then also occur from F into K. The height of tube H

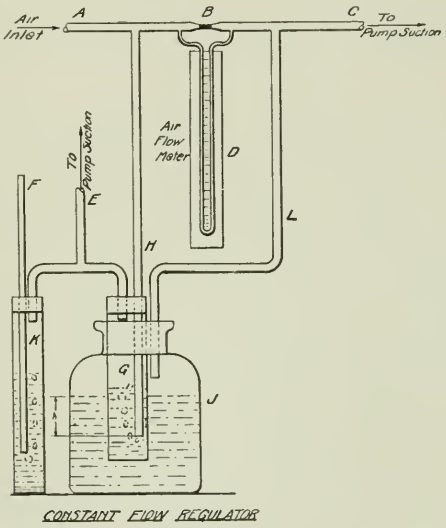

FIG. 9. CONSTANT FLOW REGULATING DEVICE FOR AIR WASHER TESTS SET

is then raised or lowered in G until flow meter D reads 1 cu. ft. per min. The drop in pressure through nozzle B is always equal to h and so long as tube H remains unchanged and a bubbling occurs then h remains constant and the flow through B will be held at 1 cu. ft. per min. The pump is then handling all air passing through B plus the waste air that enters at F and passes through H but which does not pass through the filter. Now assume that the filter resistance increases. Since h remains unchanged if a bubbling occurs in G, then 1 cu. ft. still pass through the filter but less waste will take place through F and H. So long as a bubbling occurs in G and so long as the suction at E remains constant, the volume of air passing through B and through the filter remains unchanged.

Fig. 8 shows two sets of the regulators and filters described. A single positive pressure air pump, belt driven by a ⅛ h.p. motor, is used to draw the air through the filters. Air is drawn from the washer, through the filters and regulating systems by means of a positive pressure air pump, the function of the constant flow regulators being to keep the

volume of air passing through the filters constant regardless of the varying filter resistances.

Part II of this investigation will report results of tests obtained with this apparatus on a washer handling various industrial dusts.

The writers acknowledge their indebtedness to G. W. Smith of the U. S. Bureau of Mines for valuable suggestions relative to the constant flow regulators.

DISCUSSION

J. M. ROBB: How do you get the air to go through the filter? What system or apparatus do you use to draw the air?

O. W. ARMSPACH: We use a positive pressure air pump. A pump of this kind is capable of handling a comparatively small quantity of air, but it will handle the air against a very high resistance.

S. H. KATZ: It is true that Mr. Armspach's dust determinator will eliminate a great deal of labor and time in determining dustiness. Determinations as made heretofore are difficult as well as laborious, and they require a microscope and laboratory for analytical operations. However, in order to translate resistance to flow or pressure across a filter, into terms of the weight of particles, the numbers of particles and the sizes, which are usually wanted in dust investigations such as the Bureau of Mines conducts in mines for hygiene purposes, it is necessary that some standardization or calibration of the determinator be made.

We are planning to undertake very shortly experiments in which weighed amounts of dust shall be put up in the 1000 cu. ft. gas chamber at the Pittsburgh Station. Perhaps four kinds of dust will be used, a silica dust and a limestone dust which represent rock, iron metal dust and coal dust. Of course the dusts settle continuously so that the amounts weighed represent only the dust originally suspended and dustiness will continually decrease thereafter. Samples will be taken simultaneously from that air with the sugar tube filter, the Palmer apparatus which washes air with water, and another instrument either the Hill dust counter or the konimeter which impinge dust in air against a sticky glass plate.

We shall be very glad to have Mr. Armspach join us in the tests with the Anderson-Armspach dust determinator and run it in parallel with the other instruments. The results obtained by comparison will no doubt be of considerable value.

F. PAUL ANDERSON: There is one point that might be mentioned to this body and that is this: the methods that have been devised by the physicist and the chemist for the determination of dust present many valuable merits, but there is, we think, a need for another sort of dust determinator that the engineer could use. If one man like Mr. Katz could make all the dust determinations on this planet we think it might be very satisfactory to use one of the methods that have been heretofore devised. In my opinion, the personal equation factor plays too important a part in all methods now in use to warrant any one of them as a standard field method.

The Laboratory has attempted to make a dust determinator that can be used in the field by any engineer, and the suggestion that Mr. Katz

makes that we check up our dust determinator with the methods that he has employed, will be the means, we think, of having a positive status determined for this particular suggested instrument.

In all the methods of dust investigations with which we have come in contact, we find the main difficulty to be that the dust determinations must be made by very careful workers in the Laboratory—scientists of the finest sort of training—because there is nothing more intangible that the engineer must deal with than dust. The illustration that was given to you by Mr. Armspach of fine dust created in an isolated laboratory permeating almost instantly the atmosphere of another room, indicates how intangible is dust. Therefore we are attempting to produce the means by which the engineer can in a relative way, even by a visual observation, catalog the dust in industries. He will be able to tell by the filter discoloration the probable relative amount of dust that exists at any time.

J. I. Lyle: I wish to say that this is the first instrument that I have ever seen or heard described—and we have had pretty nearly every instrument that we have heard of—that seemed to be at all practicable. I think the Research Laboratory is on the right track. I believe it will lead to the demonstration of the efficiency of machines sold for cleaning the air. I am inclined to think it is going to mean the revision of the design of many if not all air washers; because it is going to show that they do not do quite so much work as it is believed they do. We are not perfect ourselves in what we think we do, but this sort of thing rather put it up to us. We have been able previously, when they talked about the other dust counters, to sort of say, "Yes, but we haven't got the laboratory to do it in." The result is they have not been tested; there is no standard. We are on the track now of getting a standard where we can compare one dust with another or one part of one plant with another. And it not only has use in connection with air washers, but any kind of filtering mediums, which are used in a great many different kinds of work.

I think the Laboratory, with Mr. Armspach and those that have developed this, deserve a great deal of credit, and I think if they keep at it we will have an instrument that is going to be of some use to us, where laboratory instruments have not been previously.

H. M. Hart: Just for a word of information, I want to ask if it is possible to obtain cloth so uniform that it will not affect the relative readings of an instrument of this character.

O. W. Armspach: Our preliminary experiments were made with a bleached muslin filter obtained in a dry goods store. Even with this cloth where no attempt was made to obtain the best possible filter, we found that the rate of change of resistance was not affected by the slight variation in initial resistance.

Simultaneous tests made with similar pieces of cloth gave practically the same rate of increase in resistance.

H. M. Hart: One other point: Is it possible that certain kinds of dust may slide through certain kinds of cloth, where other kinds of dust might impinge and clog up the pores?

O. W. ARMSPACH: That is possible. But if you observe a filter sample under the microscope you will see large particles of dust as well as the extremely minute particles. The interstices in the cloth are many times the size of the dust particles, but it does not follow that the smaller particles pass through the filter. This is due to several causes. No doubt an electrostatic charge due to the friction of air through the filter assists in retaining the dust regardless of the size of the openings between the threads. This effect can be observed on filter cloths in the industries. The dust is often seen to form *fingers* standing out perpendicular to the plane of the cloth.

Various kinds of dust will probably have different effects on the filter. Individual allowable dust standards would therefore have to be made for the different industries.

For example, a change of resistance of 3 in. of water in 10 minutes in a paris green factory would be much more serious than a 3 in. change of resistance in a wood polishing establishment. The standards finally adopted must be consistent with the nature of the dust, with regard to the composition, size and physical characteristics of the particles.

T. A. WEAGER: I would like to ask Mr. Armspach, when the air enters an air washer, will not the dust or filter cloth which is there later become near the point of saturation?

O. W. ARMSPACH: Tests previously made by A. E. Stacey show that the resistance through a filter does not depend on relative humidity. The numerous experiments reported by Mr. Stacey demonstrated, other things being equal, that the filter resistance was practically constant for various relative humidities.

F. PAUL ANDERSON: Mr. President, in answer to Mr. Hart's question: The device presented is intended to be used for relative dust determinations. For instance, you desire to investigate dust in a particular industry. You may use almost any sort of cloth. The relative amount of resistance would be shown. The relative amount of dust under certain conditions could be determined with almost any cloth that you might pick up. You might tear up one of your wife's sheets for filters, and your relative resistances would be shown.

If you were making an investigation of dust in your particular industry you would be able to determine, we think, by a process of this kind, what the efficiency of the apparatus was under certain conditions of operation. I believe that, after all, the relative dust determinations will prove the important field for this particular instrument; because after all we have no absolute values in science; they are all relative.

In reference to the character of dust, the physicist says that some dust produces irritation in the lungs. If you started out to collect samples of dust what better sort of apparatus could you have than this, for collecting the dust to send to the chemist for analysis? It would not matter if it be metallic dust or feather dust or dust of a button factory or any other sort.

H. M. HART: My thought was not on industrial commercial washers, but other washers used in connection with ventilating systems, where

we might take the fresh air from the sidewalk intake grating in a large city. There we would get a great many different kinds of dust, and it has been proved conclusively that certain kinds of dusts go right through an air-washer, while other kinds are readily removed. The same thing might be true with this filter proposition. As far as collecting all the dust is concerned, I do not think that this instrument is the best thing. It is only good for showing relative efficiency. I think that there are other filters which would be far superior to this for collecting all the dust. The question in my mind was whether or not certain kinds of dust might not pass through the cloth with the air without affecting the pressure.

F. Paul Anderson: It is possible, Mr. Hart, to use a dozen filters if necessary. Pile them up until you are practically able to take out all the dust. But the same process of analysis would hold good: the building-up of the resistance. That is one of the first things that occurred to us. A positive air-pump that would pull against a head of 2 ft. of water would draw air through any reasonable filter.

No. 632

DUSTINESS OF THE AIR IN GRANITE-CUTTING PLANTS

By S. H. Katz and L. J. Trostel, Pittsburgh, Pa.
Non-Member

THE pneumatic tools employed in modern stone-cutting operations produce a great deal of dust, and the air, especially inside the plant is generally heavily charged with it in spite of the apparatus used for drawing off the dust. Workers exposed to stone dusts particularly those high in silica, such as granite or other hard dusts, are subject to a disease called miner's phthisis or stone-cutter's consumption. The prime cause of this disease is due to these dusts being insoluble and not readily thrown off from the lungs. Continued inhalation of dust causes an accumulation with subsequent hardening and loss of elasticity of lung tissue, and the worker becomes short-winded. Eventually, death may ensue.

Barre, Vt., is the largest granite-quarrying and cutting center in America, and is thus a fertile field for observation of granite-working trades. Recently a statistical[1] and medical[1a] study was conducted there for the purpose of learning methods of preventing disease to improve upon the course of treatment, and to determine the conditions which lead most readily to the complaint. For this purpose it was important to have information concerning the dustiness of the air. Such data are also of value for comparing dust-removing devices and suggesting means for their improvement. The U. S. Bureau of Mines, in conformation with its policy of improving health conditions in the mineral industry, undertook this investigation. The results obtained are doubtless typical of conditions existing in the entire granite industry.

ACKNOWLEDGMENTS

The investigation was conducted under the general direction of A. C. Fieldner, supervising chemist of the Pittsburgh Experiment Station, and R. R. Sayers, chief surgeon of the Bureau of Mines, to whom the writers are indebted for valuable suggestions; likewise to Frederick L. Hoffman, chief statistician of the Prudential Life Insurance Co., who lent most valuable co-operation in starting the work at Barre. Acknowledgment is also made for the cooperation and generous assistance at Barre given by William Barclay, president of Barclay Bros. Co., Joseph B. Carswell, president of Carswell-Wetmore Co., C. A. Perry, superintendent of the

[1] F. L. Hoffman, The Problem of Dust Phthisis in the Granite-Stone Industry, U. S. Bureau of Labor Statistics, Bul. No. 293 (in press).

[1a] D. C. Jarvis, A Roentgen Study of Dust Inhalation in the Granite Industry, *Am. Jour. Roentgenology*, Vol. 8, May, 1921, pp. 244-258.

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Harrison Granite Co., Robert B. Jones, superintendent of Jones Bros. Co., F. A. Grearson, president of Grearson and Lane Co., Athol R. Bell, secretary of the *Granite Manufacturers Assn.*, Dr. D. G. Jarvis, physician, and the Barre branch of the *Granite Cutters International Assn.* of America.

Many suggestive criticisms were made by the late Prof. John R. Allen and O. W. Armspach of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and Dr. R. Thiessen, microscopist of the Bureau of Mines.

CLASSES OF WORKMEN

Viewed from the amount of dust they make or breathe, there are four classes of workmen in the Barre granite-cutting establishments, namely, roughers, surfacers, finishers and muckers; however, many men do more than one class of work. Employees, such as sawyers and polishers, are not included, as they work on thoroughly wetted stones which produce no dust. These men are often separated from the others and breathe purer air.

Roughers.—Their work consists in roughly shaping and dressing down the rock as it comes from the quarry. Their tools are the old-fashioned hammer and chisel, pneumatic hammer and chisel, plug drills for boring holes, channeling tools (usually a large, flat chisel operated by an air-hammer), and the *pick,* a tool having a large number of pointed teeth set on a plane and operated by pneumatic hammer. It is a general practice for roughers to work indoors, though at one of the plants investigated, they were all outside.

Surfacers. These men (See Figs. 1 and 2) generally receive the rock from the roughers, and their work is intermediate between the latter and the finishers. The surfacers chiefly reduce the surface with large picks and multi-bladed tools operated by powerful pneumatic hammers supported above the stone on a freely moving arm of a small derrick. Lathes for shaping large columns are classed as surfacing machines.

Surfacers work indoors as well as outdoors. If the former, they are provided with exhausting apparatus, the inlet of which is made to follow the cutting tool. Fig. 1 shows a surfacer and his machine indoors: the 4-in. flexible pipe is for drawing off the dust. Vacuum systems were connected to all surfacing machines and lathes operating indoors, but in spite of them, considerable dust was thrown from the cutting machines with such force that it escaped the exhaust attachment. A surfacer operating outdoors without vacuum attachments is shown by Fig. 2, with a dense cloud of dust plainly apparent.

Finishers. These operators carve the details and delicate designs such at lettering, flowers, statuary, and other decorations. Necessarily, their eyes and nostrils are very close to the stone they cut, 8 to 12 in. being the usual distance, as shown in Fig. 3. Chisels used with small pneumatic or hand hammers are the principal tools employed in carving. Some surfaces are finished by pounding with 8- to 12-bladed hand hammers. The finishers were all found working indoors.

Muckers or Helpers. These men do the general labor around the plants, such as operating cranes, moving stones and setting them in position for the skilled workers. removing chips, and various unskilled tasks.

They are exposed to all degrees of dustiness, as their duties are not confined to any one section of the works. Thus the air they breathe over an extended period, is an average of the entire plant atmosphere.

PLANTS INVESTIGATED

Five granite-cutting plants were selected as being representative of the different types, and are hereafter referred to by Numbers 1 to 5.

FIG. 1. A SURFACER WORKING INDOORS. NOTE VACUUM ATTACHMENT FOR DRAWING OFF DUST.

FIG. 2. SURFACER WORKING OUTDOORS—NOTE DUST CLOUD.

FIG. 3. FINISHER WORKS CLOSE TO HIS JOB.

At Plant No. 1 all stone cutting was performed indoors in a single building 90 x 385 ft., with 30-ft walls, a center gable 50 ft. high, and no partitions. Only half of the floor area was used because the building had been recently enlarged.

Cutting operations were arranged lengthwise in the following order: Roughing and surfacing occupied about one-third of the floor area; finishing, the middle section, one small lathe was placed near the finishers; and the remainder was occupied by machines for sawing and polishing, everything being water-soaked around this section.

The plant was operating at half capacity or a quarter of the enlarged space capacity, and was employing about 20 men. Except in bright sun-

TABLE 1. DUST SURVEY AT PLANT NO. 1.

Sample No.	Class of Work Near Sample	Tool or Machine Used	Air Represented by Sample	Dust Content of Air: Milligrams per Cubic Meter: Over 10μ	Under 10μ	Total	Particles per Cubic Centimeter: Sugar Cube: Over 10μ	Under 10μ	Total	Konimeter: Over 10μ	Under 10μ	Total	Dust Counter: Over 10μ	Under 10μ	Total	REMARKS
1	Surfacer	Pick	Air breathed	49.1	16.2	65.3	8.	2405	2413	-	181	181	1.0	70	71	Sample taken 4 feet from tool.
2	"	"	" "	51.3	21.5	72.8	8.	1787	1795	-	389	389	0.1	13	13	" " 4 " " "
16	"	Pick	" "	79.8	34.3	114.1	10.	2543	2555	-	-	-	-	-	-	" " 3 to 6 feet from tool.
22	"	4 and 8 blade hammer	" "	56.5	18.7	75.2	8.	1437	1445	-	-	-	-	-	-	" " 3 " 4 " " "
				59.2	22.7	81.9	9.	2043	2052	-	285	285	0.6	42	43	Average
4	Roucher	Hand air pick	" "	174.0	54.5	228.5	23.	3994	4017	-	-	-	-	-	-	Sample taken 15 to 24 inches from tool.
21	"	"	" "	273.6	75.2	348.8	71.	8352	8723	-	292	292	1.6	75	77	Pick operated two-thirds of sampling time.
				225.8	64.8	288.6	47	6323	6370	-	292	292	1.6	75	77	Average
3	Finisher	Small air chisel	" "	21.4	6.1	27.5	7	251	258	-	133	133	-	-	-	Sample taken 12 inches from tool.
19	"	"	" "	77.3	39.3	116.6	12	3973	3985	0.5	399	400	0.7	19	20	Surfacer operated 20 feet away.
20	"	"	" "	31.4	12.1	43.5	4	1222	1225	-	-	-	-	-	-	Taken 10 to 12 inches from tool.
23	"	"	" "	33.2	9.2	42.4	19	1381	1400	0.7	95	96	0.2	3	3	
				40.8	16.7	57.5	11	1793	1794	0.6	209	210	0.5	11	12	Average
12	Surfacers	-	Average	6.9	0	6.9	12	692	704	-	152	152		-	-	Center of surfacing part, 2 machines operated 75 feet away.
13	"	-	"	17.4	3.9	21.3	8	1455	1463	3.0	772	775	0.1	7	7	Same position as 12 , a surfacer operated 10 feet away.
14	"	-	"	60.6	26.2	86.7	2	2698	2700	-	-	-	-	-	-	Taken 5 feet from a surfacer.
15	"	-	very dusty	541.3	159.0	700.3	80	13483	13563	6.6	1510	1517	0.7	15	16	Taken on dusty side of surfacer 6 to 10 feet away.
26	"	-	Average	5.6	2.2	7.8	8	1634	1642	0.5	103	104	-	-	-	Surfacers 40 and 60 feet away.
27	"	"	"	8.4	.4	8.8	4	865	869	-	-	-	-	-	-	Duplicate of above, but machines were idle.
				106.7	32.0	138.7	19	3455	3474	3.4	634	637	0.4	11	11	Average
6	Finishers	-	"	29.1	12.8	41.9	3	795	798	0.4	448	448	0.5	23	24	Center of building. Six finishers 10 to 15 feet away.
7	"	-	"	.6	5.3	5.9	-4	2610	2606	0.4	281	281	-	-	-	Taken in unused half opposite finishing department.
8	"	-	"	13.6	6.0	19.6	6	1608	1614	-	-	-	-	-	-	Same position as 7.
9	"	-	"	23.0	6.8	29.3	2	1321	1323	2.0	291	293	0.4	14	14	Between finishing and surfacing parts, surfacer 60 feet away.
10	Polishing	-	"	7.8	4.8	12.6	3	518	521	0.4	103	103	-	-	-	Wet all around.
11	"	-	"	9.5	5.7	15.2	6	268	274	-	-	-	-	-	-	Duplicate of 10.
23	Finishers	-	"	19.8	7.9	27.7	4	826	830	-	274	274	0.7	55	56	Center of finishing part, cutters 10 to 20 feet away.
24	"	-	"	13.6	5.5	19.1	5	1042	1047	2.0	403	405	-	-	-	Nearest finisher 15 feet.
25	"	-	"	3.7	-.3	2.9	5	710	715	0.2	141	141	-	-	-	Taken in unused half opposite finishers.
				13.4	6.0	19.4	5	1044	1047	0.9	277	278	0.5	31	32	Average
				64.3	21.3	85.6	13	2311	2324	1.5	361	363	0.6	29	30	Average for plant.

TABLE 2. DUST SURVEY AT PLANT NO. 2.

(Surfacing machines, all outside, had no vacuum attachments.)

49	*Surfacer	Pick	Air breathed	-	-	-	-	-	-	0.6	69	70	0.1	1	1	Taken 3 to 4 feet from tool, leeward side.
53	"	-	"	33.8	8.2	42.0	4	-162	-168	3.6	436	440	-	-	-	Taken about 5 feet froom tool.
54	"	-	"	72.9	13.0	85.9	5	904	909	-	-	-	-	-	-	Taken about 2 feet from tool.
55	"	-	"	23.4	7.4	30.8	5	2607	2612	0.2	19	19	0.3	3	3	Taken about 3 feet from tool.
56	"	Pick	"	47.0	10.6	57.6	5	3026	3031	-	-	-	-	-	-	Taken 3 to 4 feet from tool.
				44.3	9.8	54.1	5	1669	1674	1.5	176	177	0.2	2	2	Average
38	†Rougher	Hand hammer and chisel	"	30.4	8.9	39.3	8	925	933	1.0	206	207	0.5	6	7	Taken 3 feet from tool, 3 finishers within 20 feet.
34	†Finisher	Small air chisel	"	51.0	13.4	64.4	9	1183	1192	0.8	225	226	1.7	65	67	Sample taken 12 in. from tool; 2 other finishers in 18 feet radius.
37	"	"	"	31.3	16.0	47.3	4	1353	1367	-	389	389	5.7	91	97	" " 10 to 16 in. from tool3 others 20 feet away.
40	"	10-blade hand hammer	"	45.6	24.6	70.241		4962	5003	-	389	389	-	-	-	Two finishers in 20 feet radius.
41	"	8-blade air tool	"	24.2	14.3	38.513		2634	2647	1.5	250	252	-	-	-	Taken 14 in. from tool; one finisher 8 feet away.
42	"	Air chisel	"	69.4	44.2	113.613		7780	7793	-	474	474	0.9	26	27	Taken 8 to 13 inches from tool; 2 other finishers 12 feet awey.
58	"	"	"	59.5	13.8	73.313		3846	3859	.4	48	48	0.1	2	2	A surfacer operated 20 feet to windward.
				46.8	21.1	67.916		3626	3642	.9	296	297	2.1	46	48	Average
50	*Surfacer	Surfacer	Dusty	222.8	56.5	279.359		12254	12313	1.1	817	818	0.9	9	10	Taken 10 to 15 feet away on dusty side of machine, another operated nearby.
51	"	'	Clean	26.5	5.1	30.4-1		-9	-10	-	655	655	0.1	13	13	Taken 30 feet from above (50) machine.
52	"	"	"	22.3	4.1	26.416		410	426	-	-	-	-	-	-	Taken 40 feet next to building.
57	"	"	Average	8.1		8.1^		71	71	0.2	37	37	0.1	3	3	Taken 12 feet to leeward of surfacer.
59	"	"	Clean	7.6	1.1	8.7-1		-350	-361	1.5	17	19	-	-	-	Taken 15 feet to leeward of nearest surfacer.
				67.3	13.4	80.716		2475	2490	0.9	382	383	0.4	8	8	Average
32	†Finisher	Small air chisel	Average	6.4	2.5	8.9	5	85	90	5.0	176	181	-	16	16	Corner of room, 2 finishers 12 feet away.
33	"	"	"	11.7	10.0	21.7	4	961	965	0.3	366	366	0.7	26	27	Other corner, 1 finisher 15 feet away.
35	"	"	"	15.6	7.2	22.8	9	1702	1711	1.0	143	144	-	-	-	Center of section, 2 finishers 10 and 15 feet away.
39	"	"	"	24.1	5.8	29.912		1656	1668	0.6	232	233	-	-	-	2 carvers 5 feet away.
43	"	10-blade hammer	"	8.7	5.1	13.8 1		773	774	0.1	222	222	1.5	42	44	Finisher 6 feet away.
44	"	-	"	3.3	4.0	7.3-1		1208	1207	-	329	329	0.5	21	22	Taken in adjoining carpenter's shed, no partition, 20 feet to near finisher.
45	"	Small air chisel	"	4.8	2.1	6.9-		431	431	0.3	100	100	-	-	-	60 feet to two hear finishers.
46	"	"	"	4.4	3.0	7.4-1		1056	1055	0.2	133	133	-	-	-	Tool grinding (wet) part, 25 feet to nearest finisher.
48	"		"	2.8	1.4	4.2 3		1444	1447	-.1	196	196	-	-	-	Finisher's small room, loosely partitioned from main room,no work here when sampling.
	*Outside finside			9.0	4.6	13.6 4		1034	1038	1.0	211	212	0.9	26	27	Average.
				34.8	11.3	46.6 9		2026	2036	1.0	258	259	1.0	23	24	Average for plant.

light and in restricted places, dust was only thinly visible through the shop atmosphere.

Plant No. 2 was of the small, old-fashioned type, built in a semi-circle 120 ft. in diameter, with a boom derrick at the center, the building being 30 ft. wide. Five surfacing machines were outside, around the derrick, while indoors mostly finishing was done. Three surfaces operated at the time of this investigation, and a total of 10 stone-cutters were working; 17 cutters being the normal quota. Large doors which wall the shop on the inner side were closed at the time of this investigation, but are kept open in the summer, thus lessening the dust.

Plant No. 3 was a rectangular building 65 x 365 ft., and 80 men were working, making it a very busy place. Six surfacers worked outside and 14 inside in a portion of the shed having latticed walls, and partitioned off from the remainder. The completely enclosed section was 240 ft. long, and though a single surfacing machine was operated in this part, most of the space was devoted to finishing. One end was used exclusively for monuments and the other for building stone. Since this plant was operating at nearly full capacity it was the busiest and apparently the dustiest examined.

Plant No. 4, the largest in the district, consisted of three main buildings, one detached and idle at the time, the other two joined in T-shape. The lower section, 60 x 165 ft. and 35 ft. to the gable center, was used for stone-cutting. A quarter of the space, near the adjoining building, was used for roughing and surfacing, and the remainder for finishing. A small room on one side was partitioned off for sand-blasting. The upper section of the T was used for wet sawing, boring, and polishing, a large door connecting the two parts. Normally, 150 stone-cutters are employed here, but there were only 30 at that time.

Plant No. 5 specializes in lathe work ranging from small urns and balusters to large columns. The building was rectangular, 75 x 250 ft. and 40 ft. high. A third of the length was given to roughers and the remainder to seven lathes, of which four were operating. Wet polishing was done along one side near the lathes, and dust removal was accomplished by a vacuum attachment on all lathes. Very little dust was visible, only a slight haze through the length of the shop. Operations were at 50 per cent of capacity, 30 men being employed normally.

METHODS OF DETERMINING DUSTINESS OF AIR

The Bureau of Mines for many years has used the sugar-tube method for determining rock dust in air, and as it has long been a standard method, and since the characteristics of the sugar-tube have been determined in special investigations[2], most reliance was placed upon it in this work. Tests were also made with the konimeter and Hill dust-counter, and the results have been compared.[3] Brief descriptions of the methods are as follows:

[2] A. C. Fieldner, S. H. Katz and E. S. Longfellow, The Sugar Tube Method of Determining Rock Dust in Air, Bureau of Mines, *Technical Paper* 278. December, 1921, 42 pp.

A. C. Fieldner, S. H. Katz and E. S. Longfellow, Efficiency of the Sugar Tube Method for Determining Dust in Air, JOUR. A. S. H. & V. E., Vol. 27, March, 1921, pp. 119-123.

[3] S. H. Katz and L. J. Trostel, Comparative Tests of Air Dustiness with the Dust Counter, Konimeter and Sugar Tube, JOUR. A. S. H. & V. E., Vol. 27, July, 1921, pp. 519-528.

The sugar-tube is shown in use in Fig. 4, while Fig. 6 illustrates a recent design of a glass sugar-tube with a small recess for holding the screens which support the sugar. The method of dust determination consists in passing a measured volume of air, by means of a calibrated foot-pump, through a tube containing granulated sugar, which filters and retains the suspended dust. About 1 cu. ft. of air per minute is filtered and about 15 cu. ft. constitute a sample. The sugar-tube and included dust are then sent to a laboratory. The sugar is dissolved in water, and strained through a fine screen which removes particles above 50 microns in diameter (1 micron equals 1/25,000 in.). A small aliquot part (1 cu. mm.) is examined under a microscope at 110 diameters magnification,

FIG. 4. SUGAR TUBE IN USE SAMPLING DUST IN AIR.

and the insoluble particles of rock dust are visually divided and counted into particles above and below 10 microns in diameter. Dust in the main solution is then separated by a process of sedimentation into particles above and below 10 microns in diameter. These portions are filtered from sugar solutions on ashless filter-paper of close texture, ignited, and the non-combustible rock dust is weighed, the dustiness for the two sizes being expressed in numbers of particles per cubic centimeter of air and milligrams per cubic meter. Details of analytical technique have been described by Selvig, Osgood, and Fieldner.[4]

The dust counter shown in diagramatic form in Fig. 7 consists of a small cylinder and piston arranged for drawing a volume of air rapidly through a nozzle onto a sticky plate. Fig. 5 illustrates the dust counter with the hand case for carrying the instrument and accessories. To prepare this instrument for use, a microscope cover-glass is coated with a thin film of Canada balsam and placed in the end-piece or capsule where

[4] W. A. Selvig, F. D. Osgood and A. C. Fieldner, Collection and Examination of Rock Dust in Mine Air, Bureau of Mines Report of Investigations, May, 1920, 7 pp.

TABLE 3. DUST SURVEY AT PLANT NO. 3.

Sample No.	Class of Work Near Sample	Tool or Machine Used	Inside or Outside	Air Represented by Sample	Dust Content of Air: Milligrams Per Cubic Meter — Sugar Tubs: Over 10μ	Under 10μ	Total	Particles per Cubic Centimeter — Konimeter: Over 10μ	Under 10μ	Total	Dust Counter: Over 10μ	Under 10μ	Total	REMARKS
80	Surfacers	-	In	Air breathed	22.1	10.8	32.9	9	1041	1050	3.8	418	422	0.2	20	20	Machine in finishing department, sample taken 3 feet from tool.
85	"	-	"	"	42.1	9.6	51.7	16	703	719	-	-	-	-	-	-	Machine in surfacing department, sample taken 3 to 5 ft from tool; appears not very dusty.
89	"	-	"	"	94.7	31.2	125.9	13	4022	4035	-	518	518	0.8	9	10	Six surfacers are working in shed.
96	"	Pick	"	"	33.7	17.7	51.4	18	1968	1986	-	-	-	-	-	-	Another surfacer about 10 feet away.
91	"	-	Out	"	155.6	42.7	198.3	34	3648	3682	-	581	581	0.3	6	6	Air exhaust blew dust away from operator, wind returns some of it.
92	"	-	"	"	68.7	23.7	92.4	13	2129	2142	-	-	-	-	-	-	Two surfacers, one on each side, 15 feet away.
					48.2	17.3	65.5	14	1933	1947	3.8	468	472	0.5	14	15	Average indoors.
					112.2	33.2	145.4	24	2889	2913	-	581	581	0.3	6	6	Average outdoors.
					69.5	22.6	92.1	17	2252	2269	3.8	506	510	0.4	12	12	Average in and out.
65	Rougher	Channeling	In	"	36.4	16.3	52.7	26	4178	4204	-	300	300	-	-	-	Taken in center, 20 feet from end of building.
70	"	Hand on pick	"	"	311.0	90.0	401.0	63	12501	12564	1.5	368	370	4.9	102	107	Working on vertical surface, taken about 15 inches from tool.
73	"	Hand hammer&chisel	"	"	63.6	23.2	86.8	32	3312	3344	0.2	329	329	-	-	-	Taken 3 ft. from tool, center of section, one third length to one end.
79	"	Channeling	"	"	48.4	16.7	65.1	17	2066	2083	0.1	784	784	1.5	52	54	Taken 18 ft. from partition, surfacer 15 ft. away, 3 finishers in 20 ft. radius.
86	"	Hand hammer&chisel	"	"	47.0	10.7	57.7	14	699	703	-	362	362	-	-	-	Working in latticed wall part 20 feet from surfacers.
					101.3	31.4	132.7	30	4549	4579	0.6	429	430	3.0	77	80	Average
64	Finishers	Small air chisel	"	"	95.9	34.1	130.0	17	4860	4877	-	403	403	-	-	-	Taken 8 to 10 inches from tool edge, location 3 feet from end wall.
66	"	"	"	"	39.2	15.2	54.4	9	3683	3692	0.6	425	426	-	-	-	Taken 8 feet from end, near center, man kept stone wet to damp.
67	"	"	"	"	65.4	25.8	91.2	14	3043	3057	-	-	-	-	-	-	Taken 12 to 18 inches from tool, 30 feet from room end.
69	"	"	"	"	21.3	12.0	33.3	6	1667	1673	1.8	201	203	1.0	38	39	Taken 8 to 10 inches from tool, 100 feet from room end, one finisher 18' away.
					55.5	21.8	77.3	12	3316	3328	1.2	343	344	1.0	38	39	Average.
82	Surfacers	Surfacer	"	Average	67.9	21.4	89.3	11	1586	1597	-	666	666	1.8	89	91	7 machines operating intermittently.
84	"	"	"	Dusty	374.7	124.6	499.3	67	9733	9800	-	873	873	-	-	-	Machines 8 and 12 feet away.
90	"	"	"	Average	-	-	-	-	-	-	-	429	429	-	-	-	Machine 10 feet away.
94	"	"	"	"	35.6	10.6	46.2	7	1268	1275	-	-	-	-	-	-	Four surfacers in 40-foot radius.
95	"	"	"	Dusty	312.6	114.5	427.1	78	7939	8017	-	1199	1199	-	-	-	Taken 8 feet from 2 surfacers operating on a large stone.
93	"	"	Out	Average	112.2	36.6	148.8	24	3514	3538	-	744	744	0.5	8	9	Taken 12 feet on leeward side of surfacer.
					197.7	67.8	265.5	41	5132	5173	-	792	792	1.8	89	91	Average indoor samples.
					180.6	61.5	242.1	37	4808	4845	-	782	782	1.2	49	50	Average all.

REMARKS

All surfacers indoors have vacuum attachments.

62	Finishers	Air chisel	In	Average	43.5	18.6	62.1	15	2786	2901	1.0	435	436	0.6	14	15	Taken 6 feet from nearest finisher, 15 center from end of room, 5 finishers in 20 feet radius.
63	"	"	"	"	21.4	10.6	32.0	10	1038	1048	3.0	585	588	0.3	7	7	Taken 30 feet from end in center, 6 finishers in 20-ft. radius.
63	"	"	"	"	24.5	15.8	40.3	17	1437	1454	1.0	454	455	-	-	-	Taken 75 " " " " " 3 " " 20 " " .
71	"	"	"	"	34.6	13.7	48.3	18	2747	2765	3.2	345	348	1.2	51	52	Center of shop, sunbeam through upper windows show much dust.
72	"	"	"	"	42.4	17.7	60.1	12	2984	2996	0.6	273	274	-	-	-	Taken 6 feet from side, 100 ft. from end, 4 finishers in 20 ft. radius.
74	"	Air and handchisel	"	"	23.0	13.5	36.5	7	1865	1872	0.1	359	359	-	-	-	Taken alongside large flat building stone, 4 finishers engaged on it.
81	"	Air chisel	"	"	15.7	3.6	19.3	6	1462	1468	-	-	-	-	-	-	Taken 20 feet from end. 6 finishers in 20-ft. radius.
					29.3	13.4	42.7	12	2046	2058	1.5	409	410	0.7	24	25	Average
					83.5	28.9	112.4	21	3254	3275	1.4	502	503	1.2	36	37	Average for plant.

TABLE 4. DUST SURVEY AT PLANT NO. 4.

100	Surfacer	Pick	In	Air breathed	66.0	26.8	92.8	6	2161	2167	-	-	-	0.6	51	52	Surfacer 30 feet from end.
104	"	6-blade hammer	"	"	18.5	6.7	25.2	8	636	644	-	-	-	-	-	-	Surfacer 60 feet from end, 2 other surfacers 12 feet on either side.
106	"	12-blade hammer	"	"	14.8	2.9	17.7	11	1119	1130	-	-	-	-	-	-	Nearest surfacer 20 feet away.
					33.1	12.1	45.2	8	1305	1313	-	-	-	0.6	51	52	Average
107	Rougher	Plug drill	"	"	37.0	13.9	50.9	10	1681	1691	-	-	-	-	-	-	Taken in center of room.
108	Finisher	Small air and hand chisel	"	"	56.9	17.4	74.3		516	516	-	784	784	4.0	86	90	Two finishers are edging a large stone.
110	"	Small air chisel	"	"	148.6	54.7	203.3	16	6261	6277	-	-	-	-	-	-	Near room center.
113	"	"	"	"	31.3	15.9	47.2	9	1338	1347	-	955	955	-	-	-	Taken 10 to 18 inches from tool, 100 ft. from end, 17 ft. from wall.
117	"	Hand hammer&chisel	"	"	207.4	47.5	254.9	54	3634	3688	-	736	736	0.6	25	26	Taken 7 ft. above floor, by large high stone.
					111.0	33.9	144.9	20	2937	2957	-	825	825	2.3	56	58	Average
99	Surfacer	Surfacer	"	Average air	29.7	17.0	46.7	12	1638	1640	-	270	270	0.3	13	13	Taken 20ft. from polishing room, 15 ft. to near surfacer, dust appears light.
101	"	"	"	"	137.2	55.2	192.4	42	4125	4167	-	-	-	-	-	-	Same position as 99, but near surfacer. blows dust toward sample.
102	"	Pick	"	"	33.4	14.6	48.0	4	692	696	-	-	-	-	-	-	Surfacer 15 feet away.
106	"	Surfacer	"	"	8.5	1.7	10.2	10	523	533	-	130	130	0.2	2	2	Near center of room, surfacer 20 feet away.
					52.2	22.1	74.3	17	1742	1759	-	200	200	0.3	8	8	Average
109	Finisher	-	"	"	4.5	3.7	8.2	7	2271	2278	-	133	133	0.2	9	9	Taken 6 feet from wall, center of length, finisher 15 20 feet away.
114	'	Air chisel	"	"	18.3	8.8	27.1	8	1367	1375	-	333	333	1.0	22	23	Taken 100 feet from end, 16 feet to nearest finisher.
115	"	"	"	"	6.5	10.4	16.9	-	120	120	-	-	-	0.2	5	5	Taken 100 feet from end 8 feet from wall, 15 feet to near finisher.
116	"	"	"	"	12.9	11.6	24.5	7	2935	2942	-	344	344	0.6	29	30	Taken in center 80 feet from end, 3 finishers about 20 feet away.
121	"	Air and hand chisels	"	"	20.6	10.2	30.8	16	961	977	-	174	174	-	-	-	Taken in center 30 feet from end, one finisher 4 feet, another 10 feet away.
					12.6	8.9	21.5	8	1531	1539	-	246	246	0.5	16	17	Average
					50.1	18.8	68.9	13	1880	1893	-	429	429	0.9	27	28	Average for plant.

TABLE 5. DUST SURVEY AT PLANT NO. 5.

Sample No.	Class of Work Near Sample	Tool or Machine Used	Inside or Outside	Air Represented by Sample	Dust Content of Air: Milligrams per Cubic Meter — Sugar Tube: Over 10 μ	Under 10 μ	Total	Particles per Cubic Centimeter — Konimeter: Over 10 μ	Under 10 μ	Total	Dust Counter: Over 10 μ	Under 10 μ	Total	REMARKS
127	Surfacer	Lathe	In	Air breathed	8.1	4.2	12.3	8	438	446	-	-	-	Taken 3 feet from wall, 20 feet from end.
129	"	"	"	"	43.1	19.7	63.0	10	3712	3730	3.1	545	548	0.5	7	8	Position 4 feet from wall, 80 feet from end.
132	"	"	"	"	1.2	0.3	9.5	6	1473	1479	-	-	-	-	-	-	Near center of building.
					12.5	10.7	28.2	11	1874	1805	3.1	545	548	0.5	7	8	Average
128	Rougher	Plug Drill	"	"	10.3	5.9	16.2	12	1822	1834	-	-	-	-	-	-	Taken 30 feet from one end, no lathes operated during time of sampling.
135	"	Hand hammer and	"	"	15.8	12.6	28.4	40	2472	2512	-	440	440	2.0	36	38	Taken 40 feet from end.
137	"	" chisel	"	"	31.7	21.2	52.9	13	1466	1479	-	-	-	-	-	-	Taken 20 feet from end in center, another rougher 20 feet away.
					19.3	13.2	32.5	22	1920	1942	-	440	440	2.0	36	38	Average
124	Surfacer	Lathe	"	Average	-1.9	-1.1	-3.0	2	1589	1591	0.5	314	315	-	-	-	Taken in center 20 feet, from end; two lathes within 30 feet.
125	"	"	"	"	- .6	.2	-.4	12	619	620	-	-	-	-	-	-	Taken in center 40 feet from end; Lathe 25 feet away operated part time.
126	"	"	"	"	4.4	6.0	10.4	4	2468	2472	-	437	437	0.8	34	35	Near center, lathe 10 feet away.
130	"	"	"	"	23.9	19.5	43.4	25	7766	7791	-	322	322	-	-	-	Same position as 126; 2 lathes operating 10 feet away.
131	"	"	"	"	-1.4	2.2	.8	7	35	42	-	318	318	0.2	8	8	Center of building, lathe 15 feet away, rougher 15 feet away.
133	"	"	"	"	32.9	5.2	38.1	4	5053	5057	-	215	215	-	-	-	Near center, lathe 20 feet away, taken beside polisher.
134	Rougher	Hand hammer and chisel	"	"	- .5	3.7	3.2	-1	268	-259	-	329	329	-	-	-	Rougher 25 feet away, taken near center.
136	"	"	"	"	7.1	9.0	16.1	4	1423	427	-	-	-	0.7	21	22	Near end of building, 2 roughers 15 feet away.
					8.0	5.5	13.6	7	2212	2219	0.5	323	324	0.6	21	22	Average
					12.5	8.3	20.8	11	2077	2088	1.8	355	367	0.8	21	22	Average for plant.
					53.2	18.8	72.0	14	2383	2397	1.3	376	377	0.9	28	29	Average for five plants

TABLE 6. MISCELLANEOUS SAMPLES NOT INCLUDED IN OTHER TABLES.

36					2.7	1.6	4.3	1	554	555	0.3	47	47	-	-	-	Taken 2 to 17 minutes after work stopped at plant No. 2.
75					11.5	2.3	13.8	11	600	611	1.6	158	160	-	-	-	" 5 " 20 " " " " " " " 3.
5					10.1	1.4	11.5	4	-99	-95	-	27	27	-	-	-	" 15 " 30 " " " " " " " 1
118					3.3	52.5	55.8	1	152	153	-	119	119	-	-	-	Plant No. 4. Polishing shop, 30 feet from open door to stone-cutting shop; lamp shop
119					3.4	.9	4.3	4	463	467	-	34	34	-	-	-	" " 4 " " 100 " " " " " " " " "
103					653.7	105.6	759.3	108	6032	6140	-	858	858	2.6	78	81	Plant No. 4, Sand-blast room, very dusty, operator wears helmet.
120					1328.9	195.2	1524.1	219	12051	12270	-	1036	1036	1.2	30	31	" " 4, " " " " " " " " "
122					496.9	69.6	566.5	91	5866	5957	0.1	977	977	2.2	56	58	" " 4, " " " " " " " " "
123					445.6	71.9	517.6	70	6025	6096	0.1	1077	1077	2.1	66	67	" " 4, " " " " " " " " "
					731.3	110.3	841.6	122	7494	7616	0.1	987	987	2.0	67	69	Average for sand-blast room.
28					-1 .1	-4.2	-5.3	7	759	766	-	-	-	-	-	-	Outdoors at plant No. 1. Near wall, vacuum dust discharge, wind about 75° from line for discharge

it is supported on a small stand 1/32 (0.03125) of an inch beneath the nozzle. Three capsules may be prepared in advance for the sampling and carried in the case. The capsules are readily attached to the pump as needed by an air-tight slip-fit. One moderately rapid pull of the piston is sufficient for taking a sample in a dusty atmosphere; two, three, or as many as 10 pulls are given per sample in less dusty air. Calibration of the volume per stroke gave 5.28 cu. in. (86.5 cc.).

The cover-glasses with dust samples are permanently mounted on a microscope slide with a cloth washer or ring between to prevent contact of the dust spot with the slide, and the counting is done under a cross-ruled screen for reference lines.

A magnification of 50 diameters is used, as this is the highest power that conveniently allows the observer to bring the entire spot, approxi-

FIG. 5. DUST COUNTER WITH CARRYING CASE AND ACCESSORIES.

mately 1/16 in. diameter, into the field of vision at one time. The result of the count, multiplied by a volume factor for the instrument, gives the number of dust particles per cubic centimeter or cubic foot.

The dust counter is especially useful where permanent samples of the air dust are desired. It is also well adapted for dusts that are soluble in water or which tend to conglomerate or disintegrate in water solutions.

The circular konimeter, details of which are shown in Fig. 8, was developed in South Africa. In principle, the apparatus is similar to the dust counter, but instead of operating the piston by hand, it is propelled by a spring released by a trigger. The air passes through a nozzle 0.0225 in. in diameter, and discharges onto a circular plate glass covered with a thin coat of vaseline placed 0.02 in. (0.5 mm.) away. It then passes into a tube leading to the pump cylinder. The capacity of the pump is 0.659 cu. in. or 10.8 cc. The glass plate is cemented at its edge to a toothed brass ring which engages a worm pinion; the pinion shaft extends outside the instrument, and on it is a small cross lever; a turn of this lever brings another point on the glass plate opposite the nozzle.

The ring is numbered in sectors and 29 samples may be taken on a single plate.

To count the particles in dust spots, the glass plate is removed from the instrument and placed under a microscope at 200 magnifications, a grating being used to facilitate the counting. Two hundred diameters is employed in this case because the small diameter of the spot, about 1/40 in., allows it to come within the field of vision at the higher magnification. The results of the count are mutiplied by a factor to give the number of particles of dust in unit volume. When the glass plate is cleaned and again coated with vaseline it is ready for the next set of

TABLE 7. DUSTINESS OF AIR BREATHED BY CLASSES OF WORKMEN, AS DETERMINED BY SUGAR TUBE.

Class of workman.	Inside or outside	Number of determinations averaged.	Average dustiness of air					
			Milligrams per cubic meter			Particles per cubic centimeter.		
			Over 10μ	Under 10μ	Total	Over 10μ	Under 10μ	Total
Roughers	In	12	89.9	29.1	119.0	27	3647	3674
Surfacers	"	14	41.5	16.3	57.8	11	1818	1829
"	Out	6	66.9	17.6	84.5	11	2009	2020
"	Both	20	49.1	16.7	65.8	11	1875	1886
Finishers	In	18	61.7	23.1	84.8	14	2977	2991

TABLE 8. DUSTINESS OF AIR IN VICINITY OF DIFFERENT CLASSES OF WORKMEN.

Surfacers and Roughers	In	22	77.4	27.1	104.5	18	2996	3014
	Out	6	66.4	17.2	83.6	16	2648	2664
	Both	28	75.0	25.0	100.0	18	2922	2940
Finishers	In	30	15.7	7.8	23.5	6	1356	1362

dust samples. The advantage of the konimeter is the large number of samples that may be conveniently and speedily taken on one plate.

Order of Using Dust Instruments.—In all cases the sugar-tube samples were taken first and the dust counter and konimeter samples taken at the end of the 15-minute sugar-tube period. The two latter instruments give results of instantaneous grab samples, whereas the sugar-tube determines an average dustiness during the 15 minutes. The sampler always endeavored to have corresponding samples representative of the same conditions. Samples were taken with each instrument in various parts of the plant with the object of making a survey including all conditions of dustiness to which workmen may be exposed.

RESULTS OF DUST SURVEY

Tables 1 to 5 record the dustiness in the five plants examined, while Table 6 gives the result of miscellaneous determinations, representative

of conditions after work had stopped; a sand-blast room, and an outdoor sample near a dust discharge. Regarding the columns of Table 2, the first at the left gives serial numbers in order of sampling. Determinations of dustiness in similar places, as in the vicinity of surfacers, or air breathed by finishers, have been grouped together and averages taken. It is quite necessary to average the results of many determinations, to arrive at satisfactory conclusions because of the non-uniformity of dustiness at different points and at different times; in fact, statistical methods are necessary in dust work. The second column describes the class of work being done near the sample, which may be presumed to

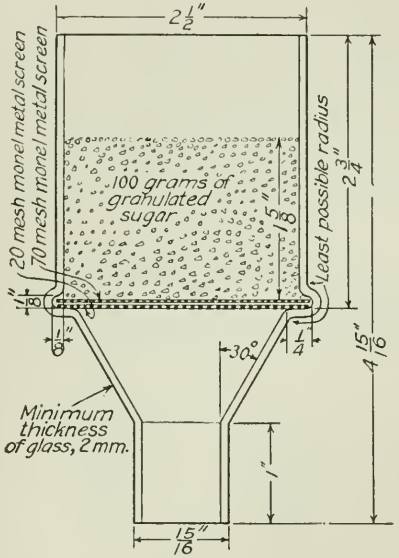

FIG. 6. GLASS SUGAR TUBE WITH RECESS FOR FIXING SCREENS IN PLACE.

cause much of the dust found. In this connection it may be noted that deposits of dust around a surfacer on clean snow outdoors, show that dust spreads in a circle of about 20 ft. radius, decreasing in quantity with the distance from the source. The finer dust is carried farther and may be borne by the air for long distances. Visual observations after work had stopped and results as shown in Table 6, indicate that about 15 minutes is required for the fine dust to disappear indoors. The disappearance must be due in a large part to the ventilation replacing dusty air with clean air, for Armspach has computed that granite dust, 2 microns in diameter, settles only 1 ft. in 16.1 minutes[5]. The third column describes the tool or machine used by the near operators, while the fourth and fifth need no comment. The sixth column gives the weights of dust in milligrams per cubic meter of air (multiplying the figure by 0.02832 gives mg. per cu. ft.). Next are columns giving numbers of particles

[5] O. W. Armspach, Theory of Dust Action, Jour. A. S. H. & V. E., Vol. 26, December, 1920, pp. 819-829.

per cubic centimeter of air as determined by the sugar-tube, konimeter, and dust counter (multiplying by 0.02832 gives millions of particles per cu. ft.). The last column indicates the point of sampling, and conditions which might influence the results. It will be noticed that a few small negative figures are given, but they have no real significance, and are simply recorded for averaging, the variance in sugar blanks is responsible for them.

VARIATIONS IN DUSTINESS

The large variations in dustiness are shown graphically in Fig. 9, which represents conditions at Plant No. 1, but which is also typical of other plants. Ordinates indicate numbers of dust particles per cubic centimeter of air, all sizes included. Ordinal numbers of samples are abscissae. The upper full line indicates dustiness in total numbers of particles as determined by sugar-tube, below it is dustiness by konimeter, and the lowest line is dustiness as indicated by the dust counter. Lack of agreement by the different methods is obvious, and this is treated in detail in subsequent paragraphs as under the heading of Comparison of Air Dust Determinations by the Sugar-Tube, Dust counter and Konimeter.

DUSTINESS AT THE DIFFERENT PLANTS

In Tables 1 to 5 the averages place the plants in the following order of dustiness, based on numbers of particles:

		Average Total Dustiness		
Order of Dustiness	Plant No.	Particles per cubic centimeter	Milligrams per cubic meter	Plant Conditions
1	3	3276	112.4	Large and busy plant.
2	1	2324	85.6	Large, working 50% capacity.
3	5	2088	20.8	Medium size, specializes on lathe turning, working 50% capacity.
4	2	2035	45.6	Small plant, working 60% capacity.
5	4	1893	68.9	Large plant, working 20% capacity.
Average for all tests		2397	72.0	

The order, based on numbers, has two exceptions from that based on weight. That this may readily occur is understood when it is taken into consideration that the particles ranged from about ½ to 50 microns in diameter, and that one particle of the large size weighs as much as one million small size. A few more or less larger particles can thus easily affect the order. The order follows the relative industry at the plants, the busiest having most dust. Of the three plants working at about half capacity, the large plant, housing surfacing machines indoors, was the most dusty, followed by a lathe plant, and that, by one where all surfacing machines were outside, a practice which undoubtedly makes less dust.

DUSTINESS OF AIR BREATHED BY DIFFERENT CLASSES OF WORKERS

The dustiness in air breathed by the different classes of workmen is indicated by Table 7. The figures represent averages of 6 to 20 determinations as stated in the third column. Roughers were found to breathe most dust, 119 mg. per cu. m. by weight and number 3674 particles per cc. They are followed by finishers who breathed 2991 particles, and then by surfacers breathing 1886 particles. Surfacers working outdoors inhaled slightly more dust than those indoors, showing that the dust removal systems were of value.

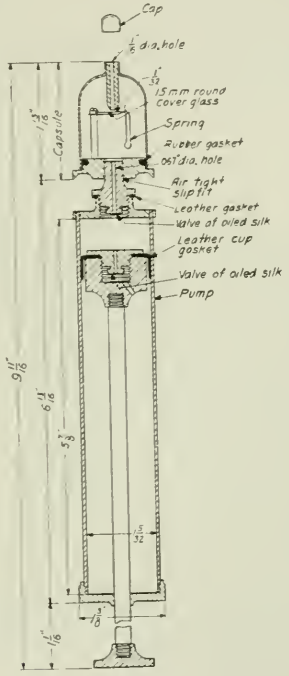

FIG. 7. DETAILS OF THE DUST-COUNTER.

The most dust was breathed by the roughers when using the small pick operated by an air-hammer: in fact, this instrument is responsible for the higher average for roughers. The pick creates excessive dust, and as the workers' nostrils are usually not more than 2 ft. from the rock, an exceedingly dusty air is inspired. Roughers breathe much less dust when using the old-fashioned hand-hammer and chisel.

DUST IN THE VICINITY OF DIFFERENT CLASSES OF WORKERS

Table 8 gives the average dustiness near the workers. Surfacers and roughers are classed together because they usually worked in proximity. The air around them carried more than twice the dust around the finishers; on the other hand, Table 7 shows that the finishers breathed more

dust than the surfacers, owing to the former operating with heads only 8 to 15 in. from their tools. The outside air near the surfacers and roughers was less dusty than the inside air, as might be expected. It will be noticed that the average air breathed by the indoor surfacers is less dusty than the average air in their vicinity. The reason for this is that the vacuum apparatus draws the dust away from the operator, and also that the exhaust air from the hammer is used to blow the dust as it is formed, away from the operator, and at the same time to clean the stone. Thus dustiness is greatest on the sides of the stone opposite the operator, and the average dustiness of air about him is greater than that which he breathes. While such use of discharged air may be good out of doors, provided the wind does not return the dust or carry it to another operator, it undoubtedly serves to increase the general dustiness indoors. Surfacers as well as others would have cleaner air if only a vacuum were used to remove dust.

SUGGESTIONS FOR REDUCING DUSTINESS

A standard of 5 mg. of dust per cu. m. of air has been proposed for the gold mines of the Rand in South Africa, as a maximum tolerable limit[6]. Although it is possible that the sugar-tube used by the Bureau of Mines retains somewhat more dust than the sugar-tube used in South Africa, nevertheless, only 6 samples in 108 were below the 5 mg. limit; and the general average of 72 mg. for five plants indicate that the existing dust conditions in granite-cutting plants, can be improved. In fact, the managements have ameliorated the dustiness by the application of vacuum systems to indoor surfacing machines, from which discharged dust accumulates in considerable bulk; and a rule of the Barre branch of the *International Granite Cutters Association of America* requires finishers to wet their work for the purpose of allaying dust; but in few instances was the practice observed. It is not the object of this paper to determine methods of dust prevention, yet in this connection it may be suggested that further adoption of wet-working methods and improvements in the application of vacuum apparatus appear as certain and practical means for betterment.

CHARACTERISTICS OF BARRE GRANITE DUST

Suspended dust in the dressing shops arises from granite rock, nevertheless it was thought that the dust might differ from the solid rock in average mineralogical composition owing to unequal friability of the rock constituents. To determine this, a sample of air-floated dust was taken from a beam 20 ft. above the floor of Plant No. 1, and was analyzed petrographically by Dr. H. Insley. The same percentage composition was found as in the original granite. A report on the petrology of several granite shop dusts is given under the heading of Petrology of Rock Dust in Barre Granite Plants, in succeeding pages.

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[6] Final Report of the Miner's Phthisis Prevention Committee. Union of South Africa, Johannesburg, January 10, 1919.

SIZES OF DUST PARTICLES

A sample of the beam-dust mentioned was analyzed for size with the following results:

Screens, Mesh Per Inch	Diameter of Openings Inches	Diameter of Openings Microns [7]	Dust Passing Through, Per Cent by Weight
20	0.0340	864	8.4 (on 20-mesh)
20	0.0340	864	91.6
48	0.0116	295	90.4
100	0.0055	140	89.2
200	0.0029	74	87.6
250	0.0024	61	57.4
325	0.0017	43	29.0

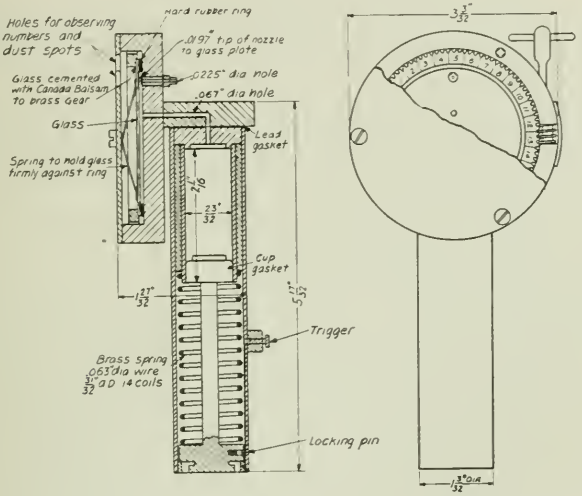
Holes for observing numbers and dust spots
hard rubber ring
.0197" tip of nozzle to glass plate
.0225" dia hole
.067" dia hole
Glass cemented with Canada Balsam to brass gear
Glass
Lead gasket
Spring to hold glass firmly against ring
Cup gasket
Trigger
Brass spring .063 dia wire 9 D 14 coils
Locking pin

FIG. 8. DETAILS OF THE KONIMETER.

The particles which were left on the 20-mesh screen were too coarse for air floating, and must have been shot from their source with enough velocity to raise them 20 ft. Of the air-floated dust, 29 per cent of the total passed through a screen of 43 microns opening. Screens for further sizing were not available.

Results of the sugar-tube samples in Tables 1 to 5 give an average of 14 particles per cc. of air, ranging from 10 to 50 microns diameter, and weighing 53.2 mg. per cu. m., similarly, those under 10 microns, microscopically visible, number 2383 and weigh 18.8. Dividing the weight of particles per cubic centimeter by their number gives the average weight per particle as follows:

Particles less than 10 microns diameter average 7.89×10^{-9} mg. per particle.

Particles more than 10 microns diameter average 3800×10^{-9} mg. per particle.

Particles all sizes up to 50 diameter average 30×10^{-9} mg. per particle.

[7] Microns ÷ 1000 = millimeters.

The weight of 7.89×10^{-9} mg. seems exceedingly small when judged by usual gross standards; billions of such particles are required to equal the weight of a pin—yet, compared with weights of the ultimate particles which constitute gas, the dust particles are enormous, being 2.4×10^{12} times as heavy as the hydrogen molecule (the hydrogen molecule weighs 3.3×10^{-21} mg.). Investigators on the Rand, South Africa[8], have determined that dust particles in a pair of silicotic lungs totaled 13 grams and numbered 20 million million particles[9], averaging 6.5×10^{-10} milligrams per particle, or about one-tenth the weight of the smaller average particle above.

SIZE OF DUST PARTICLES

Following are the average dimensions of particles, considered as cubes, determined from the weights above, and specific gravity of granite, 2.66, as average density:

1.44 microns, average diameter of particles under 10 microns,
11.27 microns, average diameter of particles over 10 microns,
2.24 microns, average diameter of all particles.

The average diameter of particles in silicotic lungs found by the South African investigators was 1.2 microns[10].

The average workman inhales 800 cc. of air with each breath, hence a granite-cutter operating in air of average dustiness inhales with each breath 1,900,000 particles, ranging from about ½ to 10 microns in diameter, which may find lodgment in the lungs. The weight, however, is so very small that it would require 100 working days of 8 hours each to inhale the air carrying to the lungs the 13 grams of dust as found in cases of silicosis. Undoubtedly only a small proportion of the dust inhaled finds even temporary lodgment in the lungs, so that it takes many years of work to develop a pronounced case of stone-cutter's consumption.

ACCURACY OF THE DUST DETERMINATIONS

Since the sharpness of separation of particles into the sizes as previously determined, affects the averages, some of the separated portions were examined microscopically by Dr. H. Insley. Particles more than 10 microns in diameter were not found among those nominally less than 10 microns, but 65 per cent of those nominally more than 10 microns were actually less. Assuming that the average small particle here is the same as that in the small fraction, it may be shown that the separation involves an error of only 0.13 per cent in the weight of large particles, or 0.37 per cent in the small. Errors from this source can be ignored.

Previous investigations by the Bureau of Mines[11] have shown that the sugar-tube, with granulated sugar, and subsequent analytical processes,

[8] The Rand is usually referred to as the Witwatersrand, a gold-mining section, situated 5000 ft. above sea level, with rocks composed of sediment fully 20,000 ft. thick. It produces 40% of the world's gold output.

[9] Final Report of the Miner's Phthisis Prevention Committee, Union of South Africa, Pretoria, January 10, 1919, p. 17.

[10] Loc. cit., p. 80.

[11] A. C. Fieldner, S. H. Katz and E. S. Longfellow, The Sugar Tube Method of Determining Rock Dust in Air, Bureau of Mines Technical Paper 278, December, 1921, 42 pp.

recovers about 90 per cent by weight of air-floated rock dust, the filtering efficiency of the sugar-tube on the number basis, about 75 per cent, is not quite so good. Also, the counting involves a personal factor on the part of the analyst, depending upon keenness of vision; the analyst counts every particle he perceives in the field viewed. Averages of five comparative counts made by three experienced analysts showed 178, 177 and 156 particles, the last being about 12 per cent below the other two. These errors of counting also affect the figures for average weights of particles.

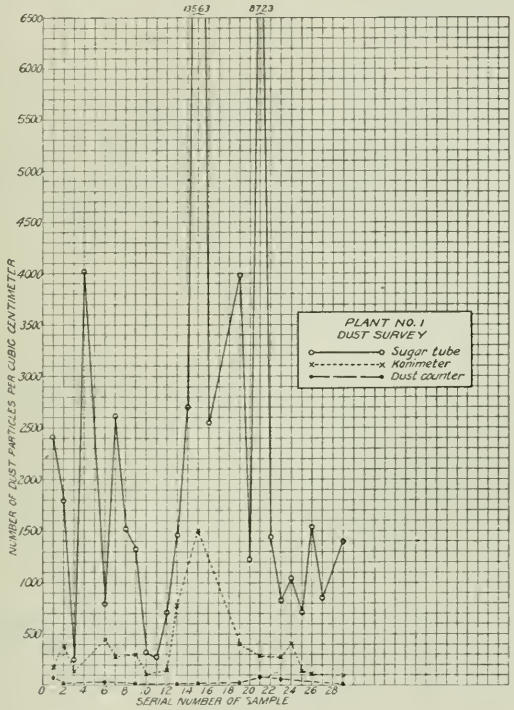

FIG. 9. CHART SHOWING VARIATIONS IN DUSTINESS OF THE AIR AT PLANT NO. 1 EXPRESSED IN TOTAL NUMBERS OF PARTICLES PER CUBIC CENTIMETER OF AIR.

Counting dust samples taken with the konimeter and dust counter, involves greater errors because the particles are close together and superimposed, making it difficult for the eye to distinguish and eliminate them as the counting proceeds. Fourteen konimeter spots were counted five times each on 9 deg. sectors by an experienced operator, and the counts averaged. They were again counted likewise on the following day, showing a maximum difference of 38 per cent, and an average difference of 16 per cent. It thus appears that the sugar-tube is most accurate, as well as most complete, because the weights of dust are obtained only with this apparatus.

COMPARISON OF AIR DUST DETERMINATION BY THE SUGAR-TUBE, DUST COUNTER, AND KONIMETER

Large variations were found in the degree of dustiness at any one plant, as indicated in Fig. 9, by total particles per cubic centimeter. However, in all but a few instances the number of particles indicated by the sugar-tube were greater than those by the konimeter, and without exception the number indicated by the konimeter were greater than those by the dust counter. The lower result obtained through the use of the dust counter is partly due to the inability to observe some of the smaller particles at the magnification of 50 diameters. On the other hand, the sugar-tube particles counted at 110 diameters were much more than those with the konimeter counted at 200 diameters.

Relations between the numbers of particles as determined by the different instruments, expressed as ratios or factors, are quite variable among sets of single tests. This is not surprising, considering the fact that nonuniformity of dustiness is apparent by gross observation with the naked eye, and certainly a large number of observations should be considered statistically for making comparisons of instruments. The ratios, then, of summations of dust particles from all comparative tests have shown the following relations:

Sugar-tube: konimeter = 6.8.
Sugar-tube: dust counter =90.4.
Konimeter: dust counter = 15.4.

The ratios of average dustiness at all plants, including all determinations in Tables 1 to 5, give slightly lower figures:

Sugar-tube: konimeter = 6.4.
Sugar-tube: dust counter = 82.6.
Konimeter: dust counter = 13.0.

The results and comparisons at each plant are shown graphically in Fig. 10, the plant averages showing fair agreement[12].

The latter three ratios agree better among themselves than the former, as the product of the two smaller ratios should equal the larger, and this is almost exactly fulfilled by the ratios 6.4, 82.6, and 13.0. For this reason they have been accepted as factors for conversion of results of dust determinations by one method into those by another.

To convert dustiness in numbers of particles as determined by dust counter to dustiness by sugar-tube, multiply by 83; konimeter to sugar-tube, by 6.4; and dust counter to konimeter, by 13. A number of samples should be taken, and results averaged to obtain reliable information, and the reliance that may be placed upon the results increases with the number of terms averaged.

SUMMARY

A survey of the dustiness of air in granite-working plants was made at Barre, Vt., and results obtained are considered representative of the

[12] These figures, slightly different, supersede those published in the preliminary report by S. H. Katz and L. J. Trostel on Comparative Tests of Air Dustiness with the Dust Counter, Konimeter and Sugar Tube, JOUR. A. S. H. & V. E., Vol. 27, July, 1921, pp. 519-528.

conditions of the industry in general, as Barre is the largest granite-dressing center in the United States. The data have been useful in connection with a medical and a statistical survey recently conducted there; and all the investigations were made for the purpose of securing information upon and reducing the conditions which lead to the disease known as stone-cutters consumption.

The workmen who dress the stone roughly into shape, were found to breathe most of the dust, totaling 119 mg. per cu. m. by weight, and numbering 3674 particles per cc. of air. They are followed by the finishers who breathe 2991 particles, and then by the operators of surfacing machines with 1886 particles. The average dustiness of five plants

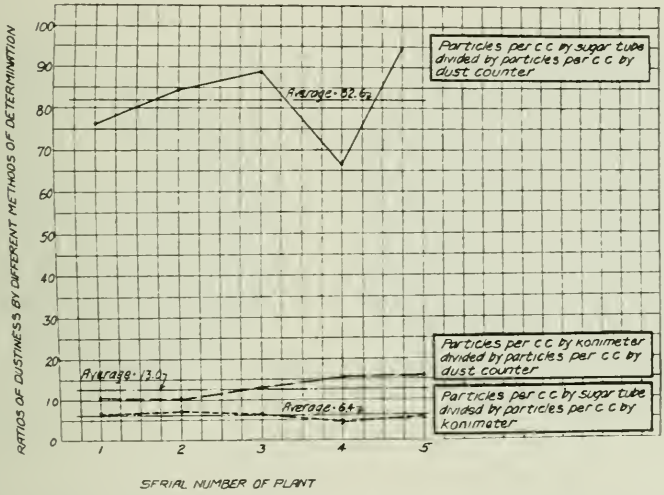

FIG. 10. CHART SHOWING RATIOS OF DUST PARTICLES PER UNIT VOLUME AS DETERMINED BY DIFFERENT INSTRUMENTS.

investigated as representative of different types, was 2397 particles per cc. weighing 72.0 mg. to the cu. m. In other words, the granite workers inhale nearly 2,000,000 microscopic dust particles with each breath. Petrographic analysis of the air-floated dust shows it to average about the same in) mineralogic composition as the granite itself.

Comparative tests of air dustiness with the dust counter, konimeter, and sugar-tube, made in the course of this investigation, have given the following factors for converting dustiness in terms of one to the other: The number of particles as determined by the konimeter multiplied by 6.4 give results in terms of the sugar-tube; results determined by the dust counter multiplied by 13 give the number of particles by the konimeter; and the number determined by the dust counter multiplied by 83 give number by sugar-tube.

PETROGRAPHY OF ROCK DUST IN BARRE GRANITE PLANTS

By Herbert Insley, Pittsburgh, Pa.
Non-Member

PETROGRAPHY OF THE GRANITE

Thin sections of Barre granite for petrographic study were made from chips collected from the coarse material drawn off by the vacuum ventilating system at one of the plants investigated. These chips were probably fairly representative of the dark Barre granite, the type of rock on which three plants were working exclusively at the time.

The essential constituents of this granite are potash-feldspar (orthoclase and microcline), quartz, biotite, and lime-soda-feldspar (oligoclase-albite to oligoclase). The accessory minerals, magnetite, titanite, allanite, etc., are not common.

The orthoclase is more or less altered and sericitized, and in places alterations were made to such a great extent that very little of the primary orthoclase remained. Microcline, on the other hand, is usually fresh and unaltered. Quartz grains show the effects of strain by the marked undulatory extinction and the pronounced cracks present. Secondary quartz is present as a cavity filling. Biotite was evidently one of the first minerals to crystallize from the molten magma, as it is found enclosed in both quartz and feldspar. The biotite is usually fresh, although sometimes slightly altered to chlorite.

In order to determine the relative amounts of the essential minerals present in the granite, a large number of the chips of the same material from which thin sections were made, were crushed, to pass through a 200-mesh screen. This fine material was thoroughly mixed, and a small portion was placed on a microscope slide and immersed in a liquid of known index of refraction. The microscope was used with a photomicrographic camera, and a plain glass plate to which tracing paper was attached, was substituted for the ground glass plate. The lens system and camera length used, gave a magnification of about 300 diameters. The images of the grains on the thin paper were traced and then cut out. The pieces of paper representing grains having an index of refraction higher than the liquid in which they were immersed, were weighed separately from those having an index lower than the liquid, and the relative percentage of the two were calculated. The index of refraction of the liquid was 1.555. Another slide on which the grains were immersed in a liquid with an index of 1.545 was prepared, and the same procedure of tracing and weighing was repeated. By observing the Becke line at the borders of the grains when immersed in the liquid with an index of 1.555, biotite could be distinguished from quartz and feldspar. When immersed in the liquid with an index of 1.545, biotite and quartz could be distinguished from feldspar. Five analyses were made with each liquid, and the average of each set calculated. The results gave the volume percentages of biotite, quartz, and feldspar (orthoclase, microcline, and plagioclase). There was no attempt made to determine the relative quantities of the different kinds of feldspar present, but in all of the analyses of the chips, as well as of the dust samples, the percentage of plagioclase was considerably less than that of orthoclase and microcline.

The volume percentages obtained were re-calculated to weight percentages by multiplying by the specific gravities of the minerals. The specific gravity of biotite was taken as 3.00, that of quartz as 2.65, and that of feldspar as 2.57. In no case were two analyses found which differed by more than 10 per cent.

This method gave the following results as applied to the chips of granite which were taken from the vacuum ventilating system at one of the plants, and were ground to pass through a 200-mesh screen:

Mineral	*Per Cent*
Feldspar	76
Quartz	15
Biotite	9

The figures were not carried beyond the decimal place, as that would

FIG. 11. APPEARANCE OF DARK BARRE GRANITE IN SECTIONS BETWEEN CROSSED NICOLS. MAGNIFICATION 15 DIAMETERS.

Composition

a—Microline.
b—Biotite.
c—Orthoclase, showing part alteration.
d—Quartz, showing undulatory extinction.

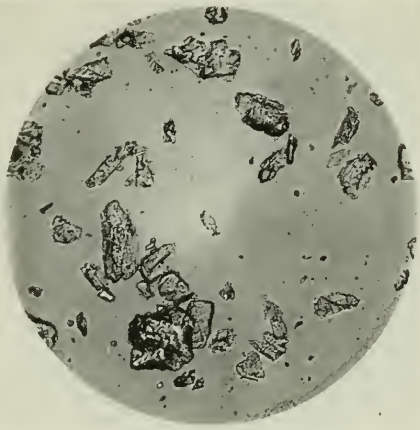

FIG. 12. THROUGH 200-MESH. FINE DUST FROM VACUUM SYSTEM DISCHARGED INTO SMALL OUTER SHED. PLANT NO. 1. MAGNIFICATION, 100 DIAMETERS.

Analysis

	Per cent
Feldspar	70
Quartz	19
Biotite, etc.	11

A rather small amount of very fine dust present. Fragments 25 microns or larger usually rounded. Smaller particles often have sharp, knife-like edges or are sometimes needle-like slivers.

probably be exceeding the accuracy of the method. By weight, the following percentages were obtained:

Mineral	*Per Cent*
Feldspar	75
Quartz	15
Biotite	10

The figure given for biotite probably includes some alteration products of feldspar whose indices of refraction are higher than 1.555, and this is

also true of all the analyses following in which the immersion method was used.

The Rosiwal method on thin sections of the chips gave the following results by weight:

Mineral	*Per Cent*
Feldspar	63
Quartz	28
Biotite	9

An estimate made by Dale[1] by the Rosiwal method on a piece of dark Barre granite gave the following results:

Mineral	*Per Cent*
Feldspar	65.522
Quartz	26.578
Mica	7.900
	100.000

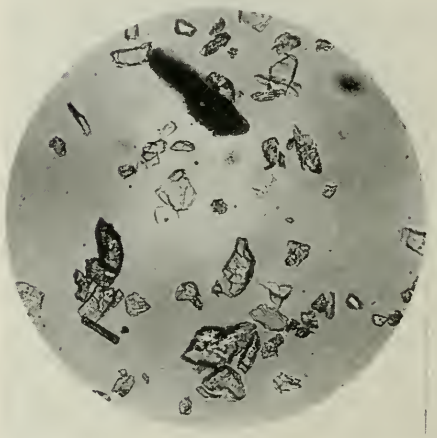

FIG. 13. THROUGH 200-MESH. FINE DUST FROM VACUUM DUST COLLECTOR, TAKEN FROM FLOOR INDOORS. PLANT NO. 1. MAGNIFICATION, 100 DIAMETERS.

Analysis

	Per cent
Feldspar	67
Quartz	22
Biotite, etc.	11

Very fine particles uncommon, others usually rounded, although sharp, knife-like particles are present. Particles of feldspar and mica usually rounded. Large flake of biotite (dark-colored), and a smaller, well-defined cleavage fragment of feldspar are visible.

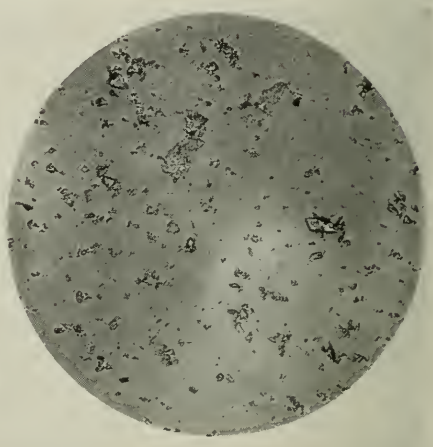

FIG. 14. THROUGH 200-MESH. SAMPLE OF FINE DUST FROM RAFTER AT PLANT NO. 2. MAGNIFICATION, 100 DIAMETERS.

Analysis

	Per cent
Feldspar	65
Quartz	21
Biotite, etc.	14

A large amount of very fine dust present, much more than shown in Figs. 12 and 13. A number of sharp, needle-like particles are present in the very fine material and can be seen very readily.

COMPOSITION OF THE DUSTS

Seven samples of rock dust obtained from various localities in the stone-cutting plants at Barre, Vt., were examined with the polarizing

[1] T. N. Dale, The Granites of Vermont, U. S. Geol. Survey Bul. 104, 1909, p. 51.

microscope, and the approximate relative amounts of each constituent (biotite, quartz, and feldspar) were determined.

For this purpose, the following simple method was used:

The sample of dust was passed through a 200-mesh screen; the product obtained was thoroughly mixed, and a small quantity was placed on a microscope slide and immersed in a liquid having an index of refraction of 1.555. The dust particles higher and lower than 1.555 were distinguished by the Becke line phenomenon and counted separately, and the percentages of each were calculated. Five fields were counted in this way and the average was taken. The same procedure was followed, using a liquid with an index of refraction of 1.545. The volume percentages thus obtained were re-calculated on the basis of the specific

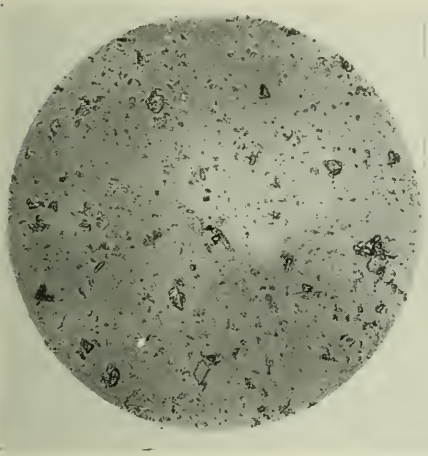

FIG. 15. THROUGH 200-MESH. FINE DUST FROM FINISHER, PLANT NO. 3. MAGNIFICATION, 100 DIAMETERS.

Analysis

	Per cent
Feldspar	67
Quartz	18
Biotite, etc.	15

Contains a rather high percentage of very fine dust. Fine, needle-like particles very abundant. Note that few opaque particles are present.

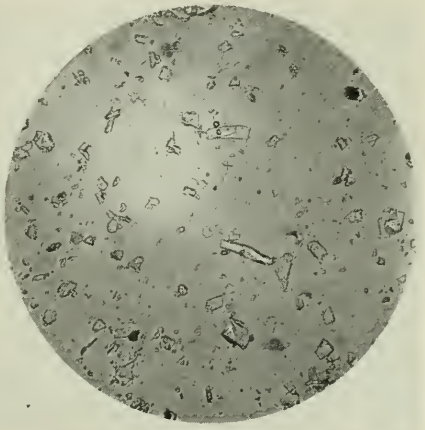

FIG. 16. THROUGH 200-MESH. AIR-FLOATED DUST FROM BEAM NEAR ROOF, PLANT NO. 1. MAGNIFICATION, 100 DIAMETERS.

Analysis

	Per cent
Feldspar	63
Quartz	22
Biotite, etc.	15

Contains a fairly high percentage of very fine dust. Needle-like particles and particles with sharp angles very abundant, especially in very fine material.

gravities of the different constituents to give weight percentages. This method was used on a portion of the sample analyzed by the tracing paper method. The results obtained by both methods were the same, although the counting method in most cases is probably much less accurate than the tracing-cloth paper method. The counting method has the advantage of being much quicker, and it probably has sufficient accuracy for the purposes of this investigation.

The mineral composition of dusts is shown under the respective photomicrographs herewith. Fig. 11 shows the appearance of granite in

section between crossed nicols. Figs. 12 to 18 inclusive represent photomicrographs of the dusts. The mineral composition of the dusts is given under the respective photomicrographs. In the descriptions therewith the term, very fine, is used to describe the particles having diameters of less than 10 microns. (A micron is 0.001 mm., or about 1/25,000 of an inch.) Considering the accuracy obtainable, the dusts appear to be about the same in composition as the granite rock.

CONCLUSIONS

Figs. 14 and 17 illustrate samples containing a large amount of very fine dust. The dust as shown in Fig. 14 was taken from a rafter, and it must have been fine enough to remain in suspension in the air for long

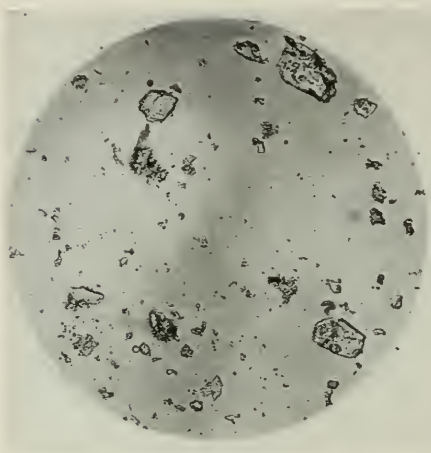

FIG. 17. THROUGH 200-MESH. FLOOR DUST FROM SURFACER. MAGNIFICATION, 100 DIAMETERS.

Analysis

	Per cent
Feldspar	59
Quartz	28
Biotite, etc.	13

Contains a very high percentage of very fine dust. Sharp, needle-like particles are fairly abundant.

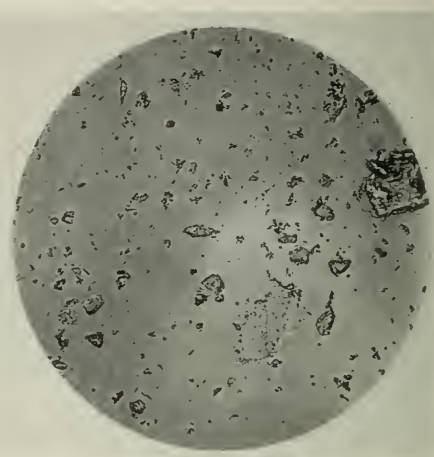

FIG. 18. THROUGH 200-MESH. PLUG-DRILL DUST, PLANT NO. 3. MAGNIFICATION, 100 DIAMETERS.

Analysis

	Per cent
Feldspar	68
Quartz	20
Biotite	12

Contains fairly high percentage of very fine dust. Sharp, needle-like particles very abundant. Opaque particles fairly common.

periods, then easily carried and lifted by air currents. The fineness of the sample as shown in Fig. 17, was probably due to the type of machine that made the dust.

It was thought before completing the mineral analyses that the fineness of the dust and the place from which the dust sample was taken, such as beams near the roof, might make considerable difference in the composition of the dust; however, the analyses as given show only small variations, and these may be due in part to errors in analysis and sampling. Whatever the difference in composition is due to, air separation of the dust constituents is so slight that it has no effect on the harmfulness of the dusts.

Silicosis or miner's phthisis is caused by the abrasion of lung tissue by fine particles of hard, insoluble dust. The properties of the rock dusts that make them injurious to lung tissue are not exactly known, but it is generally agreed that the harmfulness depends to a great extent upon the following properties: hardness of the minerals of which the dust is composed, solubility of the minerals in the fluids of the lungs, and shape and size of the dust particles. Particles less than 10 microns and more than 1 micron in diameter are thought to be more harmful than other sizes. Sharp particles with jagged edges or thin silver-like particles are probably more injurious than those with blunt edges.

Of the minerals occurring in Barre granite, only biotite, feldspar, and quartz are present in large enough quantities to be considered as possible causes of miner's phthisis.

Biotite is probably the least harmful of the three predominant mineral constituents, since its hardness on Moh's scale is only 2.5 to 3, and the particles of biotite observed in the dust samples usually had rounded edges. Feldspar is probably much more harmful than biotite, since its hardness is from 6 to 6.5, and small silver-like cleavage fragments are often present in the fine dusts. Quartz has a hardness of 7, and grains of this mineral seem to have a greater tendency to break up into particles with long, sharp edges than do the grains of biotite or feldspar. Quartz is probably the most injurious of any of the constituents of the Barre granite, but on account of the greater quantity of feldspar present, the latter mineral may play a greater part than quartz as a cause of miner's phthisis.

Particles of steel from the drills and cutting tools used in the stone-cutting plants have sometimes been considered by stone-cutters as the chief cause of miner's phthisis at Barre, but steel particles were not definitely identified in any of the dust samples. If present, they must occur in such small quantities as to be negligible as a cause for miner's phthisis.

If the relative amount of very fine dust in a dust sample is an indication of the harmfulness of that dust, then the samples shown in Figs. 14 and 17 should be most harmful; in Figs. 15, 16, and 18 should be less harmful; while those in Figs. 12 and 13 should be the least harmful.

DISCUSSION

F. Paul Anderson: This paper is a good illustration of the fine cooperation between the Bureau of Mines and the Research Laboratory of this Society. Mr. Katz is just as enthusiastic in carrying on this problem in its relation to the Laboratory as he would be if he were a member of the official force of the Laboratory; and this spirit of helpfulness exists throughout the whole Bureau of Mines. We find these scientists cooperating with the work of this Society in a very definite way.

It is unfortunate that a paper of such magnitude and importance should come at this time of day and Mr. Hallett's comments are very apropos. After all, what is the value of a meeting of this sort? The important thing is to meet each other and to develop an interest and a spirit of enthusiasm for the future study of papers that may be pre-

sented; and it seems to me that the purposes of the meeting is well defined. We cannot expect to go into an analytical discussion of any papers at the present time, but we hope that as you read these investigations at your leisure that you will find yourself enthusiastic about the work that is being done in research.

P. NICHOLLS (written): The submission of the results of the U. S. Bureau of Mines tests on dustiness in industries to this Society in itself indicates that its members are considered as the individuals who are looked to to take action, or at least to suggest remedies. It would seem advisable that the Society should formulate its viewpoints at the earliest date, and if found possible, map out a definite program for its procedure.

Wherein does the connection of the Society come? Probably in two ways, in that first it is the professional body to which has been given the care of problems connected with living conditions as far as they are concerned with the air we breathe. It would seem sometimes that we draw a distinction between individuals who live under conditions provided by the state or a public body, and those who live in conditions provided by a non-state-responsible individual. At least we plan, test, and get enthusiastically discussive about schools, auditoriums, and even offices—where we and our like have to live—but seem to fail to work up any enthusiasm when it comes to the purity of the air of a stone worker, and leave their interests to the government. Is the reason because it is considered that if the stoneworker does not like it he can do something else, whereas the child is compelled to attend school, or is it because the factory owner can talk back and the public body cannot?

Secondly, the Society has in its membership those who manufacture the apparatus necessary to remove or purify air, and is therefore interested in a financial way in extending and finding new avenues for its use, and owes it to such members to consider taking an official stand on general problems connected with their work.

It would seem that there are thus two general points of interest to the Society, first the altruistic one of giving professional advice and aid to fellow citizens, so that they may be shown their relative chance for long life and freedom from ill health under any and all conditions, or what is necessary to procure such conditions, and second, the commercial one of studying the financial economic possibilities.

It is hardly possible to establish an altruistic standard as it cannot be expressed in figures or an equation, and borders more on the philosophic, on which we may say much without establishing an agreement. On the other hand, the *financial economic*—disregarding all altruistic viewpoints, such as the benefit to the worker in better health, with increase in happiness, and possible larger earnings—can be put into figures to a large extent and involves in this case in establishing values for the yearly money costs of (1) loss in production by the factory due to poor health or inefficient work due to given conditions; (2) loss in production by the Commonwealth due to idleness from ill health, during which time also the resources of the Commonwealth are being consumed by the inactive member; (3) loss in Commonwealth financial capital due to the early death of individuals whereby the expected productive period is cut short; (4) the cost in providing apparatus or conditions which would decrease the losses due to items 1, 2 and 3. The financial economic

conditions would be those which would make the algebraic sum of all these items a minimum.

Although it may be far fetched to contemplate assembling data in such authentical definite form, would not recommendations in words tend to express the same conclusions?

The authors have so far in their two papers given us progress reports on this work without comparison of improvement in conditions produced by installation of remedial devices. Is it possible for them to give such or to indicate the program laid out for future work, the tendency of the overall analysis of the results they are obtaining, and their expectations as to what use can or should be made of their conclusions by the Commonwealth, States or this Society?

S. H. Katz[1] (written): The paper on Dustiness of Air in Granite-Cutting Plants describes the results of a preliminary inquiry into dust conditions in the Barre Granite shops. The results obtained are useful to the Bureau of Mines in its compilation of general data on dustiness in mines and shops, and the effects of wet drilling and ventilation equipment. It is believed, therefore, that the results will assist greatly in the solution of this important problem. The data presented are also of value to the Public Health Service in its study of the relation of air dustiness to disease and economic loss.

At the plants visited by the author, the managements had installed and were operating vacuum systems for dust removal that were of good design and most effective in so far as the art of dust removal in granite working plants has been developed. All indoor surfacing machines had a 4-in. flexible vacuum hose attached so that the nozzle follows the cutting tool.

The investigation being preliminary in character was not of sufficient scope to include recommendations as to better equipment, but the information obtained is sufficient to indicate that increased use of wet working methods would undoubtedly improve dust conditions, and furthermore that there is room for wide improvement in dust removing apparatus. Having the data of this paper as a basis the way is prepared for a further advance.

Regarding the action that the Society might take in this connection, the committee on Ducts and Apparatus for Dust and Fumé Removal might sometime cooperate with proper agencies representing those engaged in granite trades for the purpose of developing improved means of dust removal. And, as a commercial proposition, members of the Society might develop and install more improved vacuum dust removers at stone-cutting plants. Improved appliances might induce more extensive use of dust removers, for only the surfacers now have them.

Should the Society's members or committees bring about such action, they would at the same time, as Mr. Nicholls implies, perform a humanitarian and an economic service.

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[1] Assistant Physical Chemist, Pittsburgh Experiment Station, U. S. Bureau of Mines.

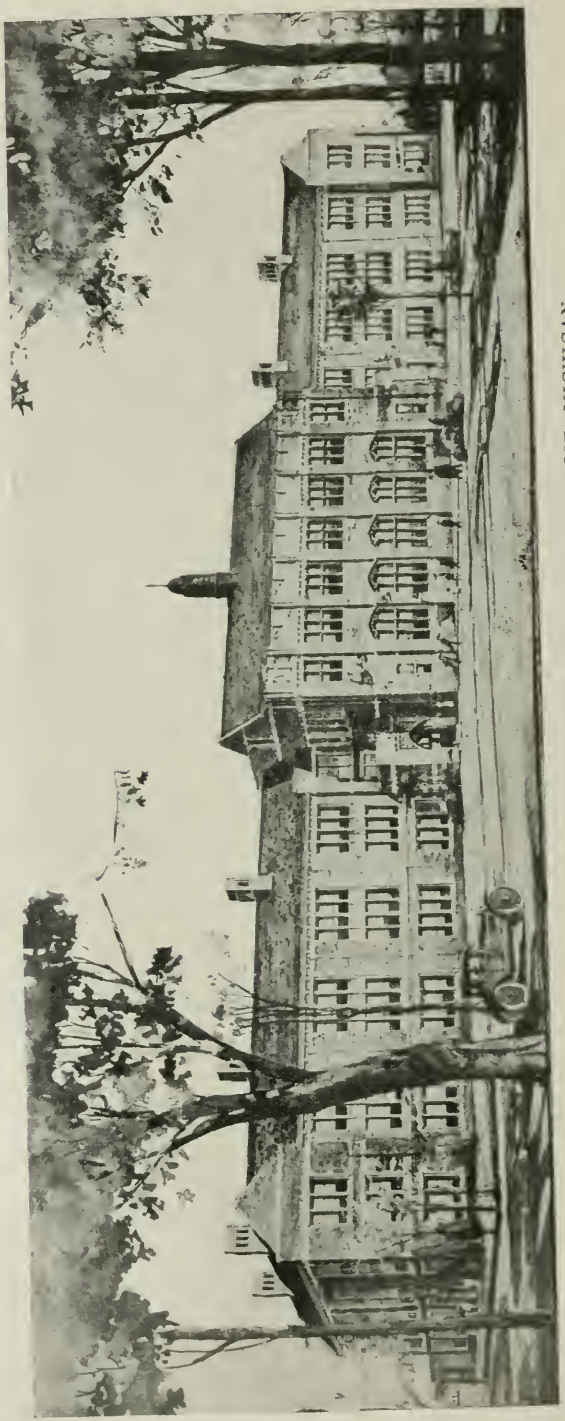


FIG. 1. LEVI M. BARBOUR INTERMEDIATE SCHOOL, DETROIT, MICHIGAN

No. 633

THE INTERMEDIATE OR JUNIOR HIGH SCHOOL IN DETROIT

By H. W. Anderson, Detroit, Mich.[1]
Non-Member

TO lay a street car line is a relatively simple engineering feat. The tracks are built level, straight, parallel, and of sufficient strength to carry well defined maximal loads. The planning and construction of the street car itself is not especially difficult. Most of the procedure has been fairly well standardized. It is easy, for instance, to determine the size of the motor required to haul a car of a certain weight, with a known hauling capacity. Long study of these problems has made them appear easy and simple.

To plan an efficient factory of almost any kind is a more difficult undertaking A study must first be made of the various operations of the factory. These must be coordinated, synchronized and constructed so as to form an orderly, efficient organization, which performs its duties without loss of time and materials. When the engineer has carefully determined the proper factory organization, the architect is called in to cooperate with him in the planning of the building which will house the factory organization. Obviously in the planning of the factory there are many more variables which must be brought under control by the engineers than in the planning of a street car track.

It has been a surprise to learn from prominent engineers that many a factory has been planned after the erection of its building, thus resulting in waste and inefficiency. The tailor who takes the measurements of the body to be clothed knows better than that. Even the school teacher is waking up to the fact that much is lost by erecting the school building before determining definitely what the school organization will be. Comparisons are odious to most people, but one will be risked. As the determination of the factory organization and program is the most important step toward the completed factory, so the determination of the school program and organization is the most important step toward the completed school. By this it is not meant to minimize the work of the architect or that of the heating and ventilating engineer. But it is a fact that the architect would be practically helpless in planning an efficient intermediate or high school building without this aid. He might construct a building on the ready-to-wear order but the chances are, by conservative

[1] Assistant Director of Educational Research, Board of Education, Detroit, Mich.

Paper presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

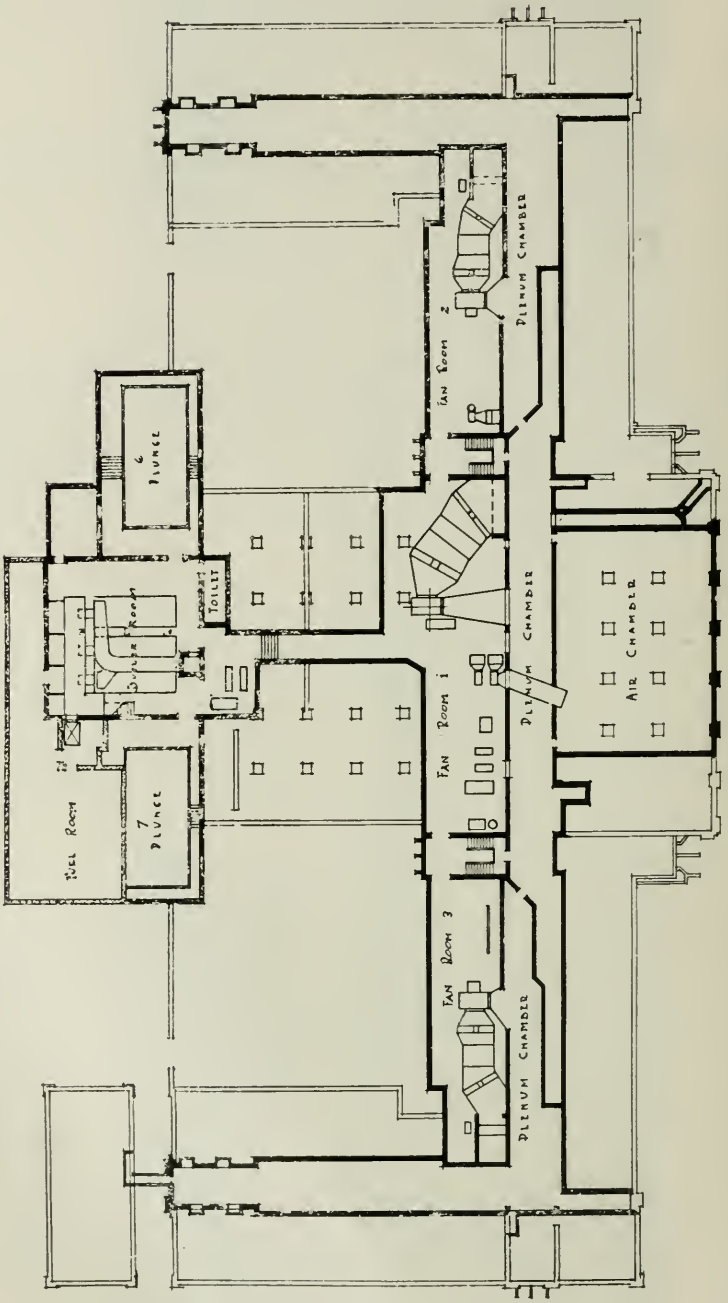


FIG. 2. BASEMENT PLAN SHOWING ARRANGEMENT OF THE HEATING AND VENTILATING EQUIPMENT.

estimate, at least 100 to 1 that it would not fit. Since the work of the heating and ventilating engineer has little relationship to the educaional program, he would undoubtedly do a good job. Table 1 shows the unbalanced use of the rooms provided for various activities in a modern high school.

TABLE 1. COMPARISON OF TOTAL CAPACITY AND ACTUAL ATTENDANCE IN VARIOUS ACTIVITIES OF A MODERN HIGH SCHOOL.

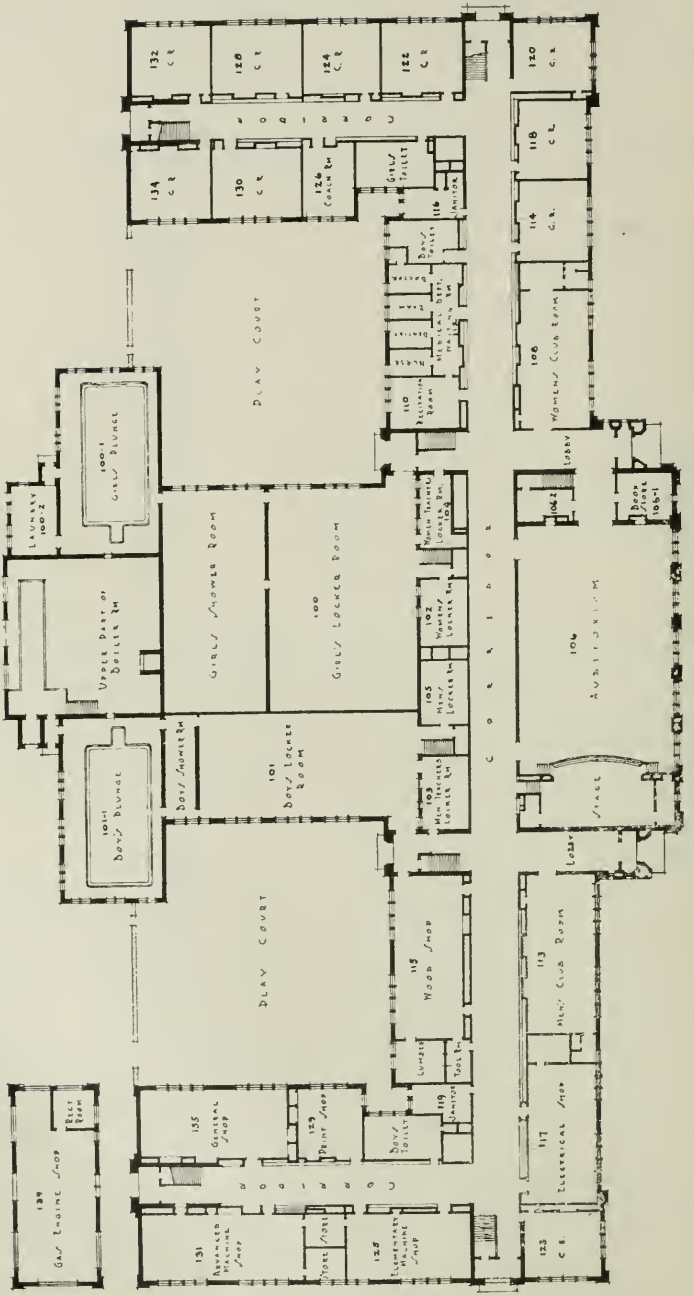
(Ten Period Day)

	Capacity	Attendance	Per Cent
Music and R. O. T. C.	....	292	..
Classrooms	7,720	6,524	85
Art	520	393	76
Library	600	457	76
Typewriting	350	254	73
Biology	300	187	62
Physical Education	1,150	631	55
Cooking	240	116	48
Chemistry	740	322	44
Bookkeeping	600	237	40
Shop	540	200	37
Sewing	600	222	37
Mechanical Drawing	620	214	35
Physics	640	136	21
Study Halls	9,470	1,747	18

Architecturally the building is a gem; educationally, an atrocity. The table shows that classrooms were used 85 per cent of capacity while study rooms, on the other hand, were used only 18 per cent. In other words some activities were seriously overcrowded while others were only slightly used. When this building was planned the architects performed a splendid service, but the educational authorities fell down on their part when they failed to give the architects a statement of their requirements, based on a scientifically worked out school program.

About three years ago Deputy Supt. Charles L. Spain and Asst. Supt. P. C. Packer, under the administration of Supt. Frank Cody, began the movement in Detroit to furnish the architect with educational specifications for schools. In planning the intermediate school, of which the Barbour, Fig. 1, and Hutchins are the best examples, extensive investigations were carried on to determine the best type of organization, the subjects pupils selected, and the probable program of the new school.

The result was the following general organization. The intermediate school day consists of six 60-minute periods with an hour for lunch. Each period provides both the recitation and the study activities under the teacher who gives instruction in the subject. During the first year the pupils' subjects are much like those of the elementary school so as to insure a gradual transition from the elementary school to the intermediate and high school organizations. When the pupil enters the eighth grade he may choose one of three courses, namely, the general, which leads ultimately to general or professional college work; the technical, which leads to engineering or technical schools; or the commercial course, which may lead directly or through college into business.



FIRST FLOOR PLAN

FIG. 3. LEFT WING OF FIRST FLOOR DEVOTED TO MECHANICAL ARTS SHOPS.

The Detroit intermediate schools are socialized schools, i. e., they aim to teach the boys and girls to become healthy, useful citizens of any American community. As far as is known the best modern thought has been utilized in providing for the health of the student. The first consideration was the plant itself. The accompanying floor plans, Figs. 2 to 6, show the layout of the school. As can be seen the building is an open type, E shaped, thus providing excellent natural light in all parts. Classrooms are located on both sides of the corridor. Lighting is unilateral, with the exception of some of the corner rooms which are intended for open air classrooms. The building is grade level, making it impossible to store children in the space rightfully belonging to coal, potatoes, and the winter's preserves. There are no wardrobes, pupils' coats and hats being taken care of in built-in corridor lockers. Toilets are conveniently located on each floor. Proper heating and ventilating mechanisms have been provided to insure healthful air.

The health training facilities consist of two gymnasiums, two covered playcourts, two swimming pools, shower baths and a medical suite. Every one of the 1800 boys and girls take an hour daily for exercise and shower bath. The floor plans, Figs. 3 and 4, show the location of the health department in the center of the building where it is easily accessible from all parts. This section of the building with the auditorium may be heated and lighted independently of the rest of the building. This provision enables the economic use of these facilities by the people of the community after school hours.

The auditorium across the corridor from the health department is in use each period. From 70 to 200 or 300 students assemble here each period to listen to lectures on social and civic affairs, and to music; to take part in community singing, dramatization, public speaking and debating. Each student has two hours a week for these activities.

The school program provides for vocational training. Since Detroit is a manufacturing city, the boys select very strongly the mechanical arts courses. For this purpose one floor of one end of the building is devoted almost exclusively to shops. Seven shops have been provided, as shown by Fig. 3, and will be used all the time by the boys of the school.

The girls' vocational work is in the home-making courses. Three cooking rooms are located on the third floor adjacent to the lunch rooms. Three sewing and millinery rooms have been located on the second floor near the rooms for art, in which designs for hats, dresses, etc., may be prepared before actually made by the students. Commercial work also constitutes an important phase of the school program. Two special rooms have been provided for bookkeeping and one for typewriting, Fig. 5. All pupils are required to take a course of subjects consisting of English, mathematics, science, history and civics, art and music.

Thus, it is seen from this brief summary that an attempt has been made to give in the Barbour Intermediate an opportunity for the best sort of training for the adolescent child who may or may not complete his schooling with the ninth grade.

This entire program was carefully worked out and estimates of the number of pupils who would take each course were prepared before any

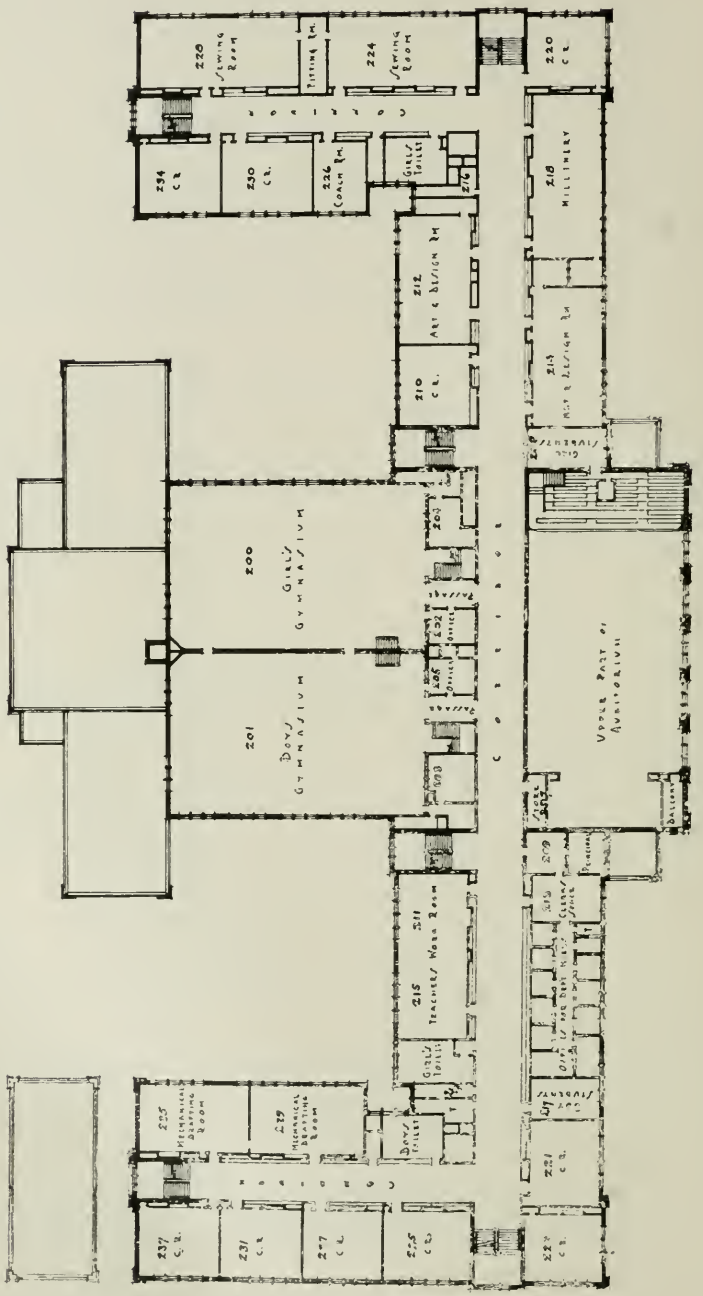


FIG. 4. SEWING, MILLINERY AND ART DESIGN ROOMS ARE LOCATED IN RIGHT WING OF SECOND FLOOR.

steps were taken to plan the building. These studies formed the scientific bases for determining the number of classrooms, shops, laboratories, etc. The result of these studies in terms of school rooms has been summarized in Table 2.

TABLE 2. NUMBER AND TYPE OF SCHOOLS ROOMS—BARBOUR INTERMEDIATE.

	No.	Size	Capacity
Class Rooms	26	22'x26'	35
General Science Laboratories	3	22'x35'	35
Music Rooms	2	22'x26'	35
Bookkeeping	2	22'x35'	35
Typewriting	1	22'x26'	35
Art	2	22'x40'	35
Sewing	3	22'x48'	35
Model Flat	1		
Cooking	3	22'x35'	24
Mechanical Drawing	2	22'x35'	35
Wood Shop	1	22'x52'	24
Print Shop	1	22'x24'	20
Advanced Machine Shop	1	22'x52'	24
Electrical Machine Shop	1	22'x35'	24
Electric Shop	1	22'x43'	35
General Shop	1	22'x46'	35
Auto Shop	1	26'x52'	35
Gymnasiums	2	50'x80'	105
Corrective Gymnasiums	2	16'x45'	35
Swimming Pools	2	25'x40'	35
Play Courts	2	44'x50'	70
Tutoring Rooms	4	17'x22'	24
Home Science	1	22'x26'	24

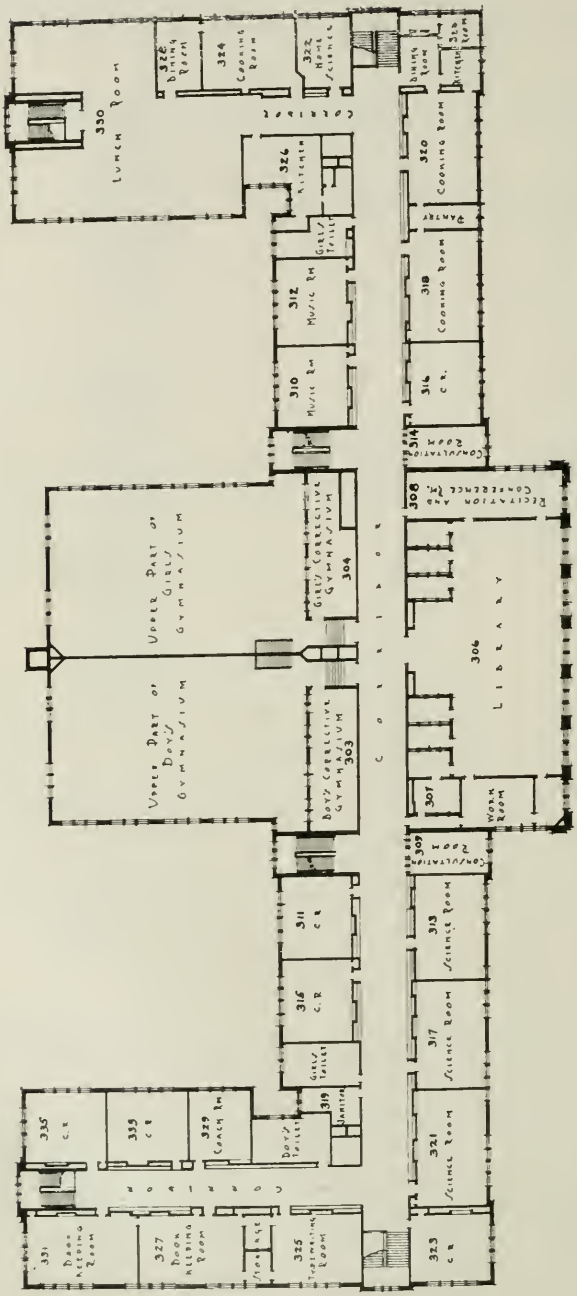
The special administrative, community and non-instructional rooms of the building have been summarized in Table 3.

TABLE 3. SUMMARY OF THE NUMBER OF SPECIAL NON-INSTRUCTIONAL ROOMS.

	No.		No.
Toilets	18	Locker Rooms (gymnasium)	6
Offices (administration, etc.)	15	Shower Bath Rooms	2
Janitor's Rooms	5	Medical Clinic	1
Cafeteria	1	Conference and Consultation	4
Boy Scouts	1	Auditorium	1
Community	2	Heating and Power Unit	1

This Society is doubtless particularly interested in the heating and ventilating equipment used in the new intermediate schools, as these represent the latest standards for all new Detroit schools.

For several years, prior to the building of these schools, the consulting engineers for the Detroit Board of Education have been using what might be called a modified split system, wherein sufficient direct radiation was supplied to take care of approximately 75 per cent of the heating, and the air used in ventilation warmed sufficiently above room temperatures to supply the necessary additional heating effect. The temperature of the air delivered to each room was under individual thermostatic control, there being separate ducts from the fan to each room. This system, by introducing air into the room considerably above room temperatures,



THIRD FLOOR PLAN

FIG. 5. ROOMS FOR COMMERCIAL AND DOMESTIC SCIENCE COURSES STUDENTS ARE ON THE THIRD FLOOR.

utilized to a large degree the stratification of the air and its settling as it cooled for purposes of distribution.

During the war period the cost of sheet metal duct work mounted so that it became necessary to design a system which would give at least equal results and which could be installed at lower first cost. With this problem to be solved, the consulting engineers, of the Board of Education, gave the matter intensive study and finally decided that a projection system, pure and simple, in which the air would be distributed by virtue of its velocity was the proper solution to work on. They had used ceiling outlets for air delivery many years ago and it was decided to come back to this system in a more refined way, provided the architects could help solve the question of installation of proper ducts in the fireproof buildings. The cooperation of the architects in using the steel pan type of concrete floor construction simplified the problem, making it possible to utilize the void spaces between the concrete floor joists for ducts, these being laid out for the proper location.

The system as worked out has proved very simple and cheap. This was partly made possible by the fact that the Detroit Board of Education started to build schools without basements, Detroit being one of the last of the large cities to adopt this policy. It is therefore possible to utilize a basement corridor under the first floor corridor, as a large distributing plenum chamber for tempered air. This chamber is made of sufficient height for one to walk through and is equipped with hose connections and floor drains so that it can be easily washed down and kept sanitary. It also serves as a space for carrying steam and return lines, electric conduits, etc. The use of this large distributing plenum chamber has eliminated practically all the sheet metal work formerly used, there remaining simply the verticals to connect with the ceiling ducts.

The results obtained from Detroit's latest system with ceiling distribution of air have proved very satisfactory not only as to first cost, but in the efficiency and uniformity of air distribution. Although the systems are laid out for the old standard of 30 cu. ft. of fresh air per minute per pupil, it is expected, due to the uniformity of distribution, to cut this amount down to at least 20 cu. ft. of air per minute and possibly to 15. Furthermore, it is possible to recirculate any amount of the air, inasmuch as the fresh air is drawn from above the roof with a connection from the shaft to the attic, which, with pneumatically controlled dampers, makes it possible to use all recirculated air or all fresh air or any desired percentage of each.

The fresh air supply for the building is taken in through three vertical shafts extending from the basement to a point above the main roof. Two supply fans with a capacity of 45,000 cu. ft. per minute each (Fig. 8), and one with a capacity of 75,000 cu. ft. per minute (Fig. 7) deliver the air from these shafts to the basement air chamber, extending completely around the building. Each supply fan apparatus is equipped with tempering coils, air washer and reheater coils with automatic temperature and humidity controlling devices.

Vertical galvanized iron risers conduct the air from the air tunnel to the various rooms. These risers are installed in the corridor walls and connect to short horizontal ducts in the floor construction or attic above

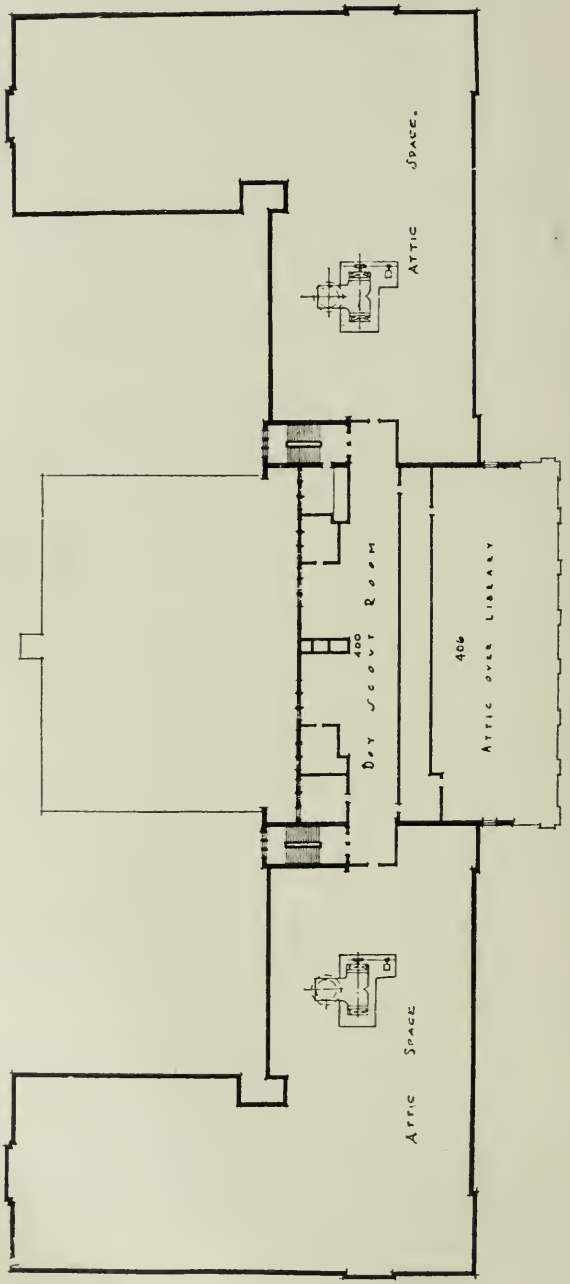


FIG. 6. TWO MOTOR-DRIVEN EXHAUST FANS INSTALLED IN ATTIC OR FOURTH FLOOR.

the respective rooms. Two of these horizontal ducts are provided for each standard class room and in each duct are installed three special diffusers or a total of six per classroom, symmetrically arranged in the ceiling. Each classroom is provided with two vent openings for the egress of vitiated air, located at the floor on the corridor side, equipped with grilles or registers, and connections to vertical metal flues leading to the attic. The vertical supply and vent flues in the furred corridor wall are so arranged with respect to the corridor wardrobe lockers, classroom bookcases and closets that the floor space taken up by these flues is reduced to a minimum. The short horizontal ducts in the floor con-

FIG. 7. SUPPLY FAN APPARATUS FOR AUDITORIUM AND GYMNASIUM SECTION OF BUILDING.

struction leading from the risers to the outlets in the various rooms are incorporated in the concrete construction, the duct occupying the space ordinarily taken up by one row of steel or terra cotta floor tile. In the Barbour School removable steel tile forms were employed in the construction of the concrete floors and in this instance the void spaces formed by the removal of the steel tile were utilized as ducts, a single sheet of galvanized iron and the usual ceiling lath and plaster forming the bottom of these air passages to which the diffusers are attached.

All vent flues from the various rooms open to the attic and as a supplement to the supply fans, two motor-driven exhaust fans of the propeller type are provided in the attic for exhausting the air from the attic through roof ventilators.

The ceiling diffusers are set behind pressed metal grilles flush with the plaster, and are equipped with two sets of diffusing vanes at right angles to each other, as shown in Fig. 9. This arrangement of vanes is designed to diffuse the air in the shape of a cone from the outlet to the floor of the room. The outlets in the ceiling are so spaced that these cones of air intermingle and overlap at the level of the breathing zone. Smoke tests made in rooms equipped with this system of ventilation showed conclusively that the air distribution is very uniform. Relatively low velocities at the outlets are employed in order to avoid uncomfortable drafts from

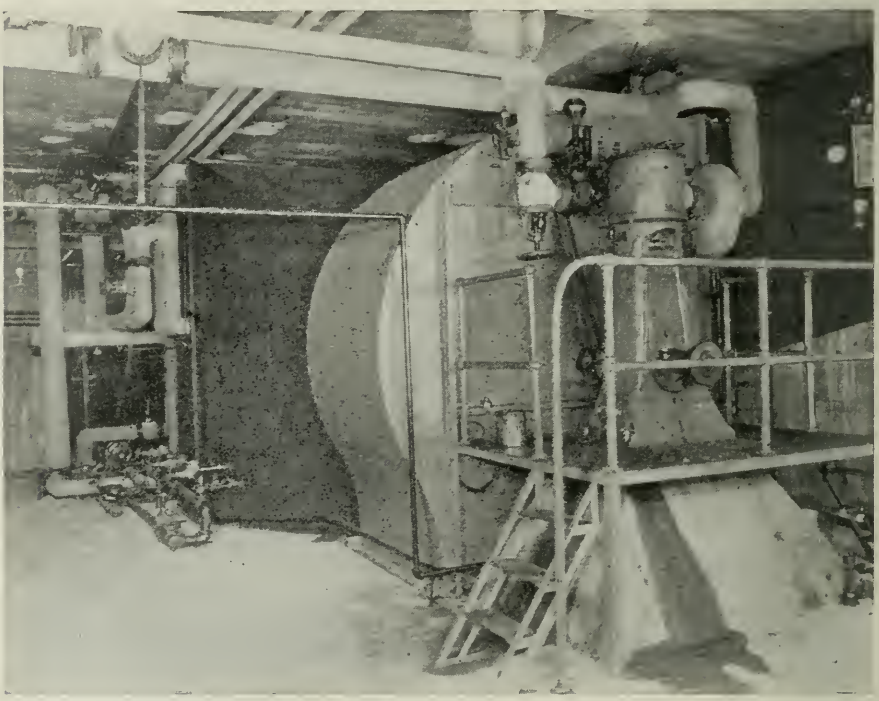


FIG. 8. TYPICAL SUPPLY FAN APPARATUS FOR ACADEMIC SECTION OF BUILDING.

the air supplied directly overhead. Fig. 10 shows the typical arrangement of ceiling outlets and vent registers in the classrooms.

The auditorium, gymnasiums, swimming pools, locker rooms, shower rooms, club rooms, etc., are located in the center portion of the building. These rooms constitute that portion of the building used for community purposes and are ventilated by means of the larger supply fan and so arranged that both the heating and ventilating systems for this portion can be operated separately from the remainder of the building.

The auditorium is provided with a reversible system of ventilation. During the cold weather the warm, fresh air supply is admitted through four large grilles located in the corners of the room. The vitiated air is drawn off through mushroom ventilators located in the floor under the

seats by means of an exhaust fan, which delivers this air through roof ventilators. The space under the auditorium floor is finished off to form an air chamber, which can either be connected to the exhaust fan or to the supply fan by means of suitable transfer dampers. During the warm weather cool air is admitted to this chamber from the supply fan and forced through the mushroom ventilators in the floor, the vented air

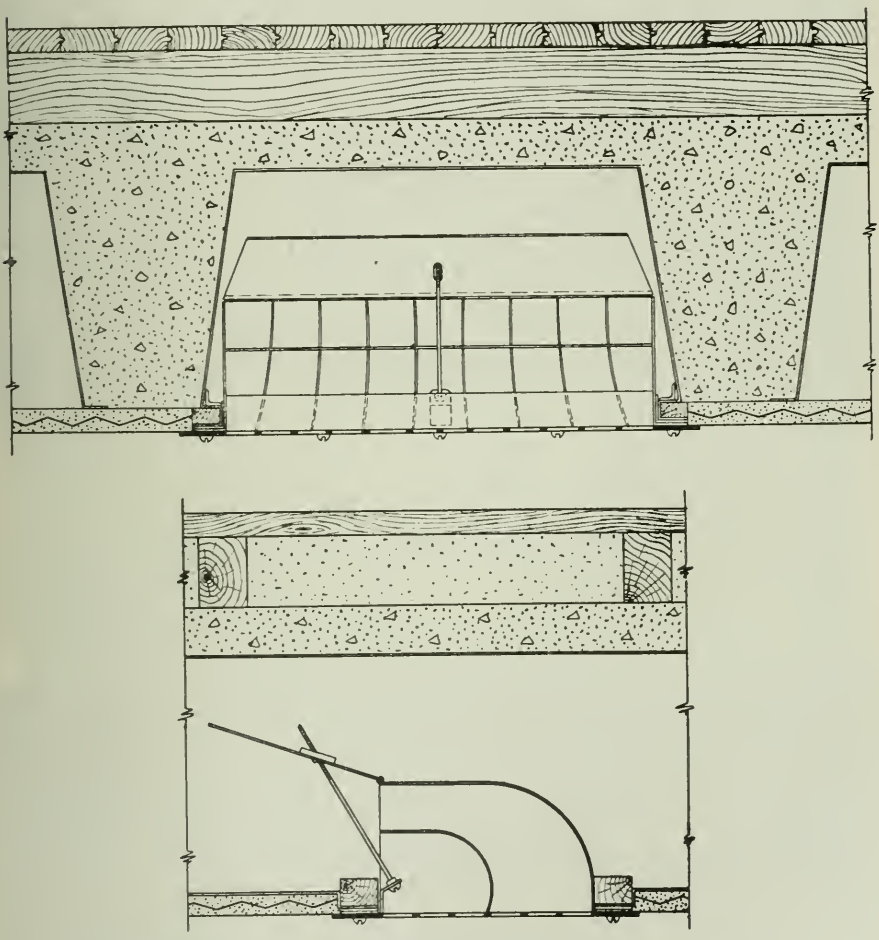


FIG. 9. DETAIL SHOWING CONSTRUCTION OF SPECIAL CEILING DIFFUSERS.

leaving the auditorium through ornamental plaster grilles in the ceiling connected with vent flues leading to roof ventilators. The entire system of transfer dampers required to reverse this ventilating system is controlled pneumatically by means of diaphragm motors with operating switches located in the fan room. The air supply to the gymnasium is admitted through diffusers and grilles located immediately below the window sills and the vent openings for winter operation are located at the

FIG. 10. TYPICAL CLASSROOM INTERIOR.

four corners of the room at the floor. During the warm weather, vent openings in the roof connected to the roof ventilators are used.

The fresh air supplied to the locker rooms is admitted through ceiling diffusers and a fan exhaust system is provided, together with a system of underground vitrified clay tile ducts, which exhausts the air from the room through the individual clothing lockers, serving the double purpose

FIG. 11. HAIR DRYER.

of ventilating the room and the lockers. This arrangement has proved very satisfactory as a method for preventing the odors from lockers contaminating the air in the rooms.

A special feature of the girls' locker room is a new type of hair drying equipment. Due to the relatively large number of girl students using the shower rooms daily, it was necessary to provide some means for the rapid

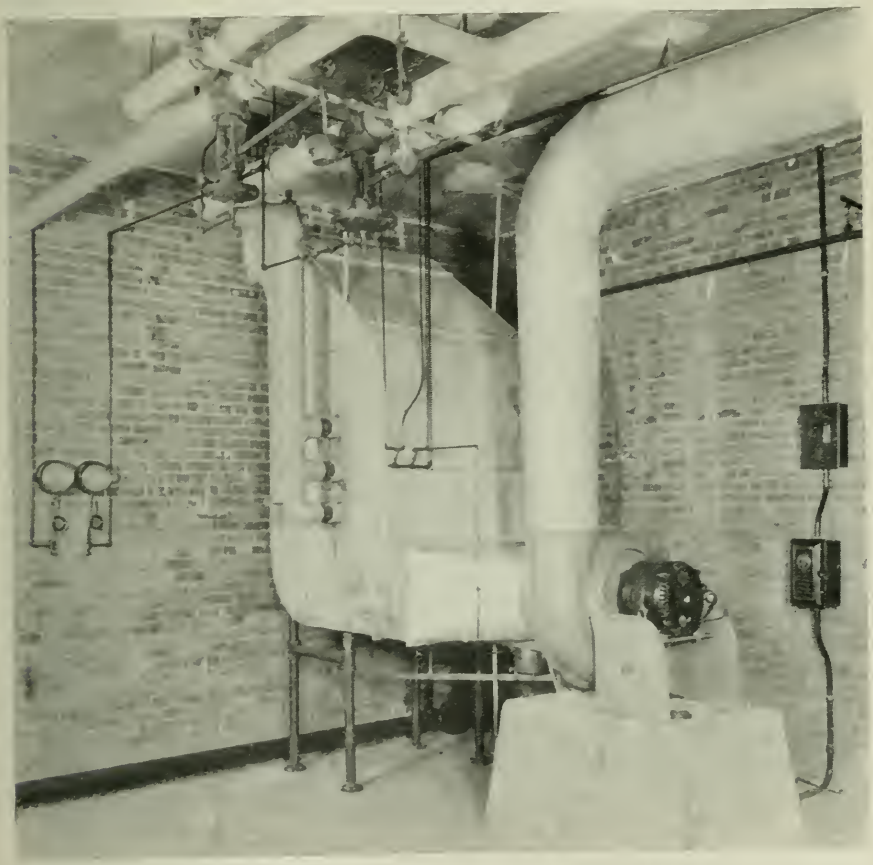


FIG. 12. HAIR DRYER SUPPLY FAN APPARATUS.

drying of the hair. A view of this apparatus, which is installed in the girls' locker room, is shown in Fig. 11. This is constructed of sheet metal, suitably reinforced, and is equipped with 24 adjustable funnel shaped outlets, the whole being finished in hard white enamel. The outlets can be readily turned to any desired angle to accommodate students standing or sitting and are so designed that when turned to the extreme lower position, the air supply is automatically cut off from the outlet. The hair dryer is supplied by a small fan (Fig. 12) located in the fan room below the locker rooms, drawing washed air from the plenum chamber through heater coils under thermostatic control. The motor

driving this fan is equipped with an automatic starter and is operated by means of suitable push buttons located in the locker room adjacent to the hair drying apparatus. This equipment will accommodate approximately 100 pupils per hour.

All toilet rooms are ventilated by means of an exhaust system, the two fans used for this purpose being located in the basement fan rooms, delivering the air through roof ventilators. The kitchen, cafeteria, chemical laboratories and domestic science rooms are equipped with separate exhaust systems, similar to that provided for the toilet rooms.

FIG. 13. BOILER ROOM.

The direct radiation in general is located around the exterior walls beneath the windows. All radiators are equipped with metal shields (Fig. 10) to protect the pupils seated nearest the radiators from the radiant heat. All temperature regulation is accomplished by controlling the direct radiation, the temperature of air supplied to the rooms being maintained constant at approximately room temperature. Diaphragm motors are used for controlling all dampers in connection with the ventilating system. These are operated by means of pneumatic switches placed in the fan rooms and the air supply for the same is taken from the compressed air system provided in connection with the temperature regulation equipment.

The boiler plant is located at the rear of the building with a floor level approximately 10 feet below grade. The fuel room floor is at the same

level and the coal is admitted directly from the trucks through manholes into the fuel storage. A 1000 lb. capacity electric platform hoist is provided for elevating the coal in charging barrows from the coal room level to the firing platform, which is approximately 10 ft. above the boiler room floor. This hoist is also used to elevate the ashes from the boiler room floor level to grade. Three 150 hp. return tubular boilers with dutch oven settings and natural draft automatic stokers are installed for providing steam to the plant. The firing is automatic, the coal being dumped directly from the elevated firing platform, Fig. 13, to the tops of the dutch oven furnaces of the stokers.

The large main supply fans are driven by low pressure engines and exhaust through oil separators into the heating system. The boiler plant is designed to operate at approximately 40 lb. pressure and this pressure is used for operating the fan engines. The fan coils and direct heating system are operated at low pressure by means of pressure regulating valves.

DISCUSSION

President McColl: This paper is open for discussion now. Mr. Anderson is willing to answer any questions asked him.

H. M. Hart: At what velocity is the air introduced through ceiling registers into classrooms?

President McColl: The velocity of entering air is about 90 ft. per minute. That rate was established experimentally as follows: Some of us with rather thin hair on top stood under the outlets to determine how high a velocity we could have without feeling uncomfortable. No discomfort was experienced with an air velocity of 90 ft. per minute, so we designed the system with that limit.

W. H. Carrier: Have any tests been made with higher velocities or to reduce the effective velocity a short distance from the outlet?

President McColl: The air is given out in a pyramid with an angle of about 30 deg. each way from the vertical. The velocity leaving the outlet is higher than 90 ft. but the average velocity from the outlet to the floor is about 90 ft. per minute.

Mr. Carrier: Does that include the effect of the diffusion?

President McColl: Yes.

Mr. Hart: Do I understand that the 90 ft. per minute flow is not the outlet velocity at the register?

President McColl: Your assumption is correct, Mr. Hart.

Mr. Hart: My question was, what is the velocity from the register?

J. W. Snyder: The velocity of air is 250 ft. per minute from the register.

Mr. Carrier: Was there any test made as to what velocities occured at the breathing level of the room, if the average velocity was 90 ft. per minute?

President McColl: No test was made.

MR. CARRIER: Was the anemometer used?

PRESIDENT MCCOLL: No, you would have to use smoke, as in Dr. Hill's test.

MR. CARRIER: I think that it would be a good place to use the hot-wire type of anemometer for determining velocities in various portions of the room.

MR. HART: What temperature of air is used in the hair dryer?

MR. SNYDER: About 100 to 120 deg. fahr.

PRESIDENT MCCOLL: The capacity of the hair dryer in the Barbour School is 150 girls per hour, although the paper said 100 girls. When the girls have bobbed hair, the capacity is increased.

MR. HART: What velocity of air is used in the hair dryer?

MR. SNYDER: The volume is 100 cu. ft. per minute through each outlet. The funnels are 4 in. in diameter, at the outlet and 3 in. in diameter at the throat.

MR. MEYER: Do you ever attempt to use but one boiler and dispense with standby equipment? In Minneapolis we have designed quite a number of schools and for economy, put in one boiler, to operate at 120 per cent of boiler rating, at maximum demand and satisfactory operation has resulted. I wonder what your practice is here. Do you install duplicate boiler equipment?

PRESIDENT MCCOLL: Always.

MR. MEYER: After determining maximum steam demand at minimum temperature do you select the boiler to operate at normal rating or at overloads? You design for zero here, don't you?

PRESIDENT MCCOLL: Yes, and at 140 per cent rating.

MR. MEYER: Do you use stokers?

PRESIDENT MCCOLL: Yes.

No. 634

THE CRITICAL VELOCITY OF STEAM IN ONE-PIPE SYSTEMS

By F. E. Giesecke[1], Austin, Texas
Member

THE term, one-pipe system, is used here to designate a steam heating system in which the steam flows in one direction and the condensate in the opposite direction through the same pipe and at the same time. The system shown in Fig. 1 is an example of a small one-pipe type.

A one-pipe heating system functions properly when the steam flows with so low a velocity that the current of steam has no effect upon the

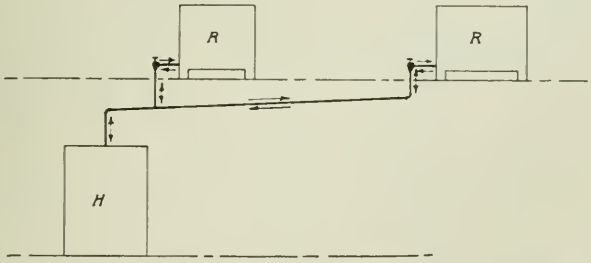


FIG. 1. A SMALL ONE-PIPE HEATING SYSTEM.

flow of the condensate and the two opposing currents flow with uniform velocities.

The maximum velocity of the steam which permits the uniform flow of the condensate in a one-pipe system is the *critical velocity*.

When the velocity of the steam exceeds the critical velocity a part or all of the condensate will be held back, temporarily or permanently, in the radiators or in the larger pipes by the current of steam, and the flow of the condensate will consequently be irregular or intermittent, or will cease entirely.

To determine the critical velocity of steam in various pipes and valves, the writer conducted two series of investigations; one in 1911 at the Texas Agricultural & Mechanical College and the other in 1922 at the University of Texas. The actual observations in the first series were made by J. E. Baade, a senior student, and those in the second series were made by

[1] Head of Engineering Research Division, University of Texas.

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

J. M. Garrett, a graduate student, and by W. M. Wilson, a junior student. All three men deserve much credit for their careful and painstaking work.

Both series of tests were made with practically the same equipment, but improvements in the apparatus and in the method of testing were made from time to time during the progress of the investigations.

In the first series steam at a pressure of 4 in. of water only was used; in the second series steam at pressures of 4 in., 12 in., and 20 in., was employed.

Fig. 2 shows the apparatus used during the latter part of the second series of tests; it consists essentially of a valve *A*, through which steam is supplied; a pipe *P*, for which the critical velocity of the steam is to be determined; and a condenser *R*, of variable capacity, by means of which the velocity of the steam in the pipe *P* could be regulated.

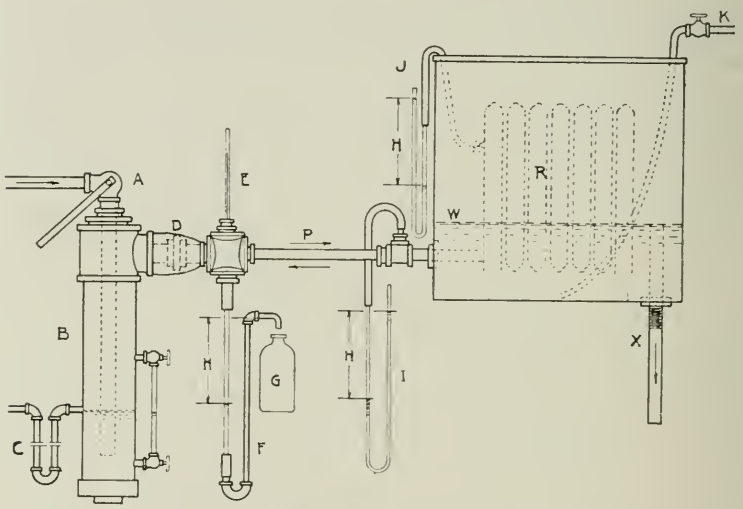


FIG. 2. APPARATUS FOR DETERMINING THE CRITICAL VELOCITY OF STEAM.

Described in detail, the method of conducting the tests was the following: Steam at 150 lb. pressure was passed through two pressure reducing valves and then through a hand-operated modulating valve *A*, into a steam separator *B*; the separator consisted of a 5-in. pipe with an overflow *C* and a 1½-in. pipe through which the steam entered the separator in such a way as to pass through a body of water in order to remove also any superheat which the steam might have. From the separator the steam passed through a well insulated connection *D*, to the pipe *P*, to be tested, and through this pipe to the radiator *R*, in which it was condensed. The condensate flowed through the pipe *P*, into the manometer tube *F*, and through that tube into a vessel *G*, of known capacity. By measuring the time necessary to fill the vessel *G*, the velocity of the steam in the pipe *P*, could be calculated.

The radiator *R*, was located within a galvanized iron tank and partly immersed in running water; the water was admitted to the tank through the pipe *K*, and the attached hose, and was discharged through the pipe *X*.

The discharge pipe had a long and easy fitting thread so that the pipe could be turned by hand and the level of the water varied at pleasure. A half turn of the pipe X, for example, would change the level of the water about 1/16 in. and would make a slight change in the quantity of steam condensed in a unit of time and a correspondingly small change in the velocity of the steam in the pipe P.

In order to determine how the steam pressure varied in the apparatus, two additional manometer tubes were attached as shown in the figure; in all cases the three manometer tubes showed practically the same pressure; this was to be expected since the drop in pressure in the pipe P, for the low velocity of the steam is practically negligible.

In conducting a test, the first run was made with the water level, W, very low; additional runs were then made, each one with the water level slightly higher than during the preceding run. The time necessary to

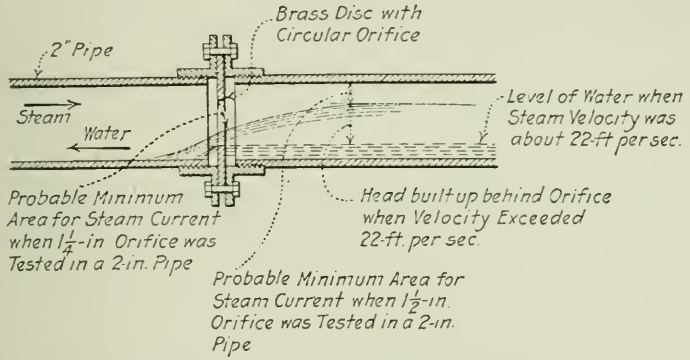

Brass Disc with Circular Orifice
2" Pipe
Steam
Water
Level of Water when Steam Velocity was about 22-ft per sec.
Probable Minimum Area for Steam Current when 1¼-in Orifice was Tested in a 2-in. Pipe
Head built up behind Orifice when Velocity Exceeded 22-ft. per sec.
Probable Minimum Area for Steam Current when 1½-in. Orifice was Tested in a 2-in. Pipe

FIG. 3. PROBABLE FLOW OF CONDENSATE DURING TESTS FOR CRITICAL VELOCITIES FOR CIRCULAR ORIFICES

fill the vessel G was noted for each run and the shortest observed time corresponded to the highest velocity, i. e., to the critical velocity, since when the time necessary to fill the vessel increased, although the water level W was raised, some of the condensate was evidently held back by the current of steam.

The following readings in minutes and seconds were taken in the test to determine the critical velocity in a 1-in. black pipe, 12 in. long and sloping 5/32 in. in 12 in. with steam at a pressure of 12-in. of water: 7:00; 6:45; 6:35; 6:30; 6:25; 6:18; 6:10; 6:12; 6:12; 6:15. The shortest time was 6:10 and corresponded to a velocity of 25.4 ft. per second.

When the shortest time had apparently been determined, the level of the water in the tank was lowered considerably so that the excess water in the radiator would drain out. The water level was then raised very slowly while additional test runs were being made and the shortest time redetermined. This process was repeated until several determinations of the same value for the critical velocity had been secured.

The critical velocity of 25.4 ft. per second, cited above, was calculated as follows: The vessel contained 991.7 cc. of water or at 212 deg. fahr. 2.1 lb. of water. The flow of steam through the pipe P was therefore 0.0057 lb. of steam per second or at a pressure of 12-in. of water, 0.148

cu. ft. per second. The cross-sectional area of the pipe was, by measurements, 0.842 sq. in. and hence the velocity of the steam, 25.4 ft. per second.

This value is not, however, the actual maximum velocity of the steam but only a theoretical value from which the maximum quantity of steam flowing through the pipe can be calculated by assuming the entire pipe to be filled with steam. In reality only a part of the pipe is filled with steam and the remainder with water and the actual maximum velocity of the steam is therefore higher than the critical velocity calculated above.

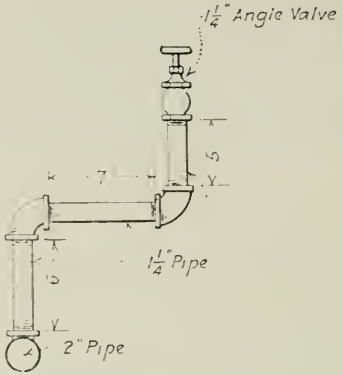


FIG. 4. RADIATOR CONNECTION WITH ANGLE VALVE, AS TESTED.

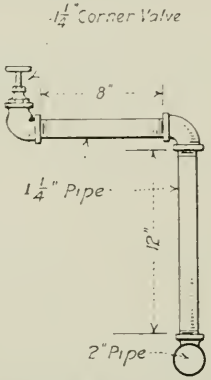


FIG. 5. RADIATOR CONNECTION WITH CORNER VALVE, AS TESTED.

The difference between these velocities may be considerable. For example, in the case of the circular orifices in vertical brass plates it was quite evident from the stains on the brass plates that the current of condensate occupied a considerable portion of the cross-sectional area of the orifice and that therefore the maximum velocity of the steam through the contracted opening in the orifice was considerably greater than the value calculated for the critical velocity and shown in the accompanying table.

By the method outlined above the critical velocity of steam has been determined for short horizontal pipes, short vertical pipes, circular orifices in thin brass plates in vertical and horizontal positions, angle radiator

TABLE 1. CRITICAL VELOCITIES OF STEAM IN PIPES, ORIFICES, VALVES, AND RADIATOR CONNECTIONS

Type	Diam.	Pitch Inches per Foot	Pressure Inches of Water 4	12	20	Average Velocity	B.t.u. per hour with 4 inch Pressure	Square feet of Radiation at 250 B.t.u. per square ft.
Horizontal pipe, 12 inches long	1	1/32	22.2	22.4	22.3	22.3	17,600	70
Horizontal pipe, 12 inches long	1¼	1/32	22.5	22.8	22.6	22.6	30,900	123
Horizontal pipe, 12 inches long	1½	1/32	23.2	23.7	23.3	23.4	43,400	173
Horizontal pipe, 12 inches long	1	1/8	24.8	24.5	24.6	24.6	19,400	77
Horizontal pipe, 12 inches long	1¼	1/8	25.1	25.1	24.9	25.0	34,200	136
Horizontal pipe, 12 inches long	1½	1/8	27.4	27.0	27.2	27.2	50,500	202
Horizontal pipe, 12 inches long	1	5/32	25.6	25.4	25.6	25.5	20,100	80
Horizontal pipe, 12 inches long	1	1/4	27.0	26.8	26.8	26.9	21,200	84
Horizontal pipe, 12 inches long	1¼	1/4	27.1	27.2	26.8	27.0	36,900	159
Horizontal pipe, 12 inches long	1½	1/4	29.2	29.4	29.1	29.2	54,100	216
Vertical pipe, 12 inches long	1		30.5	30.4	30.8	30.6	24,200	96
Vertical pipe, 12 inches long	1¼		32.3	31.8	31.7	31.9	43,600	174
Vertical pipe, 12 inches long	1½		34.2	35.9	35.6	35.2	65,400	261
Orifice in thin vertical sheet	¾		22.9	22.1	22.2	22.4	9,000	36
Orifice in thin vertical sheet	1		22.5	22.2	21.9	22.2	15,900	63
Orifice in thin vertical sheet	1¼		29.3	29.2	28.1	28.9	32,800	131
Orifice in thin vertical sheet	1½		25.0	24.7	25.5	25.1	40,400	161
Orifices in thin horizontal sheet	¾		23.2	22.8	22.6	22.9	9,200	36
Orifices in thin horizontal sheet	1		23.4	22.7	22.9	23.0	16,500	66
Orifices in thin horizontal sheet	1¼		27.4	26.8	25.8	26.7	30,200	120
Orifices in thin horizontal sheet	1½		22.7	22.8	22.7	22.7	36,600	146
Angle radiator valve	1		21.4	20.8	21.1	21.1	15,100	60
Angle radiator valve	1¼		22.1	21.8	22.4	22.1	25,000	100
Angle radiator valve	1½		19.3	19.0	19.8	19.4	31,200	124
Corner Radiator Valve	1¼		20.0	19.8	19.4	19.7	22,400	89
Radiator connection with angle valve, Fig. 4.	1¼		18.3*	18.8*	18.9*	18.7*	25,600	102
			22.1†	22.6†	22.7†	22.5†		
Radiator connection with corner valve, Fig. 5.	1¼		16.7*	16.5*	16.4*	16.5*	22,600	90
			20.1†	19.8†	19.7†	19.9†		

* Based on diameter of pipe.
† Based on diameter of opening in valve.
Orifice sizes are actual sizes.
Pipe sizes are nominal size.

valves, corner radiator valves, and the ordinary short radiator connections employing short vertical and horizontal pipes and either angle or corner valves.

The results of these determinations are shown in the following table. The critical velocities in this table were calculated for the actual internal diameters of the pipes and for the actual openings in the valves.

An examination of this table shows that:

a. The critical velocity is practically independent of the steam pressure.
b. The critical velocity increases slightly with the diameter of the pipe.
c. The critical velocity is decidedly higher for vertical than for horizontal pipes.
d. The critical velocity in nominally horizontal pipes increases with the slope of the pipe.
e. There is no definite relation between the critical velocities in orifices and those in short pipes.
f. Considerably less steam can flow through an angle valve than through a short pipe of the same nominal size partly because the critical velocity is lower in the angle valve than in the pipe and partly because the opening in the valve is smaller than that in the pipe.
g. Less steam can flow through a corner valve than through an angle valve.
h. The capacity of the ordinary radiator connection is determined by the capacity of the radiator valve.

The peculiar fact that the critical velocity for the 1½-in. orifice was lower than that for the 1¼-in. orifice is explained as follows: The orifices tested were in thin brass plates which were clamped in 2-in. flange unions; a drain cock was attached to the pipe beyond the union and by means of it it was found that so long as the velocity of the steam did not exceed about 22 ft. the surface of the water beyond the brass plate was practically even with the lower edge of the orifice but that when the velocity of the steam increased above 22 ft., a head of water was built up behind the orifice and that this head increased with the velocity of the steam.

It is practically certain that the critical velocity of the steam for the 1¼-in. orifice was determined by the cross-sectional area between the surface of the water and the upper edge of the orifice whereas for the 1½-in. orifice it was determined by the cross sectional area between the upper surface of the water and the upper surface of the 2-in. pipe as shown in Fig. 3. Hence if the 1½-in. orifice had been tested in a 3-in. pipe instead of in a 2-in. pipe, a higher critical velocity would have been found.

In examining the table it should be noted that the number of B.t.u. shown is based on a steam pressure of 4 in. of water and that the number of B.t.u. transmitted increases slightly as the steam pressure increases; for example, if the pressure is increased from 4 in. to 5 lb. gauge, the number of B.t.u. is increased about 25 per cent.

The table shows clearly that a considerable economy would be effected if the manufacturers of radiator valves increased the size of the openings in the valves so that as much steam can pass through the same as through a short vertical pipe in a one-pipe system.

In conclusion the writer wishes to acknowledge the cordial cooperation and assistance accorded him in this investigation by A. M. Seiders, Supt. of the Power Plant; Prof. H. C. Weaver and F. Morris of the Department of Mechanical Engineering of the University of Texas.

No. 635

CAPACITIES OF STEAM HEATING MAINS AS AFFECTED BY CRITICAL VELOCITIES OF STEAM AND CONDENSATE MIXTURES

Report of Cooperative Work by A. S. H.-V.E. Laboratory and the U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

By F. C. Houghten, Pittsburgh, Pa. (Member)
and
L. Ebin, Pittsburgh, Pa. (Non-Member)

IT is the scope of this paper to consider only one phase of the general subject of the flow of steam in pipes; namely, the critical velocity of steam and condensate mixtures.

The flow of steam in pipes has been the object of a great many investigations in the past, both practical and theoretical. These investigations have resulted in a large number of formulæ and expressions giving relationships between weight of steam flowing, density of steam, pressure drop, and diameter of the pipe. If we were to judge from the volume of literature on this problem, we should infer that no further investigation was necessary. The phase of the subject in hand, however, has received very little attention and very little appears in the literature on the problem. This may be for the reason that the subject does not lend itself so well to theoretical and mathematical analysis as does the flow of steam alone, but requires a great deal of experimentation, much of which requires special apparatus and equipment.

James A. Donnelly [1] to whom the writers are indebted for many helpful suggestions and criticisms has long studied this phase of the problem of steam flow and has often recommended the need of research along this line. Mr. Baldwin and R. C. Taggart [2] in discussing Mr. Donnelly's paper give an account of some experiments with critical velocities. Mr. Baldwin also recommends some safe velocities for use in various pipes. F. W. Raynes [3] gives a formula for the height of wave produced in the returning condensate in a horizontal pipe. Harding

[1] James A. Donnelly, Standard Sizes of Steam Mains, Vol. 13. Transactions A. S. H. & V. E., 1907, p. 43.

[2] Discussion Carrying Capacity of Pipes in Low Pressure Steam Heating, Vol. 13. Transactions A. S. H. & V. E., pp. 64-71.

[3] Raynes, Heating Systems, 2nd ed., p. 259.

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

and Willard [4] recognize the importance of keeping below the critical velocity in boiler outlets and in one pipe risers and recommend safe velocities for these connections.

The Research Laboratory has for some time had this problem under consideration, but has been unable to undertake the work until recently, for want of available room with suitable facilities.

In March, 1922, arrangements were made with the United States Bureau of Mines, whereby space was made available for this work in the Fuels Laboratory of the Pittsburgh Station. Since that time the apparatus shown in Fig. 1 and Fig. 2 has been assembled and a number of experiments conducted which give promise of a successful series of tests.

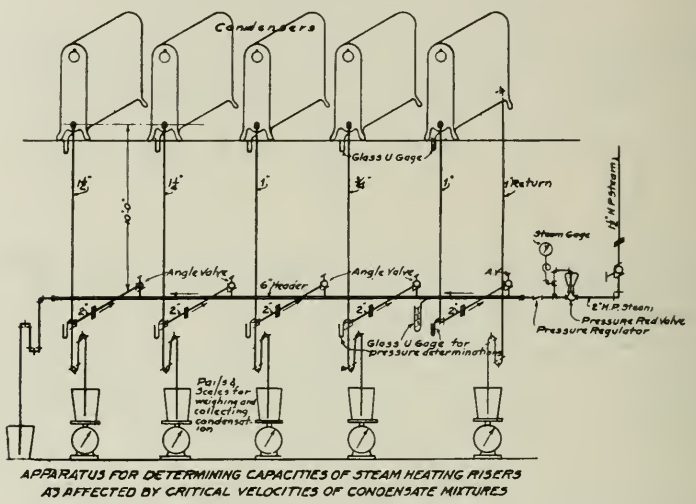

Condensers
Glass U Gage
Return
Steam Gage
Angle Valve
Angle Valve
Pressure Regulator
Glass U Gage for pressure determination
Paired Scales for weighing and collecting condensation
APPARATUS FOR DETERMINING CAPACITIES OF STEAM HEATING RISERS AS AFFECTED BY CRITICAL VELOCITIES OF CONDENSATE MIXTURES

FIG. 1. ARRANGEMENT OF EXPERIMENTAL APPARATUS

By the term critical velocity, as used in connection with steam and condensate mixtures, is meant the velocity at which the flow of steam and the counter flow of condensed water in the same pipe seriously interfere with each other. This use of the term critical velocity must not be confused with the term critical velocity as used in the various branches of engineering as meaning, the velocity at which stream line flow of any fluid, under given conditions, ceases and turbulent flow begins.

The critical velocity of steam and condensate mixtures has many applications in steam heating. In some cases where it is desired to carry the water along with the steam it is desirable to so size the pipe that the velocity of the steam is above the critical point. Often, however, it is desired to have the water of condensation flow counter to the steam. In such cases it is necessary to so size the pipe that the velocity of the steam is below the critical point.

The plans to be carried out in connection with the outline of this problem include the determination of the critical velocity in each of the following cases:

[4] Harding & Willard, Mechanical Equipment of Buildings, Vol. 1, p. 229.

THE PROBLEM

Mains pitching back toward the boiler; both one and two pipe systems.
Riser connections pitching back to the main; both one and two pipe systems.
Horizontal offsets in steam risers, both one and two pipe systems.
Risers—both one and two pipe systems.
First floor radiator connections.
One pipe radiator connections.

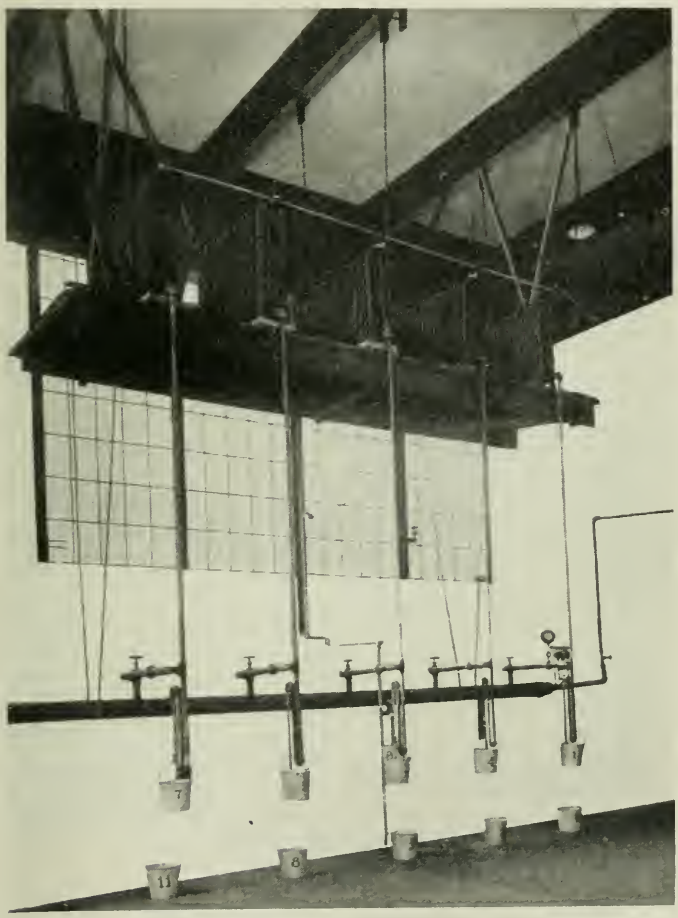


FIG. 2. TESTING APPARATUS FOR DETERMINING THE CRITICAL VELOCITY IN RISERS

Level pipes, all lengths.
Horizontal pipes, all degrees of pitch and length.
Globe valves in vertical and side positions.
Angle valves for various lifts of the disc.

The effect of rounded entrance, reamed, and unreamed, pipe, also the effect of pipe cut with one and three-wheel cutters.

Influence of the starting load on the critical velocity, also the effect of the size of the air valve orifice on the starting load.

At the present time the critical velocity in risers is being investigated. For this work the apparatus shown in Fig. 1 and Fig. 2 has been built and a number of tests have been made. The apparatus consists of a 6-in. header and five risers; one ¾-in., two 1-in., one 1¼-in., and one 1½-in. Each riser is connected to the header through a 2-in. angle valve, three 2-in. ells, a 2-in. union and 30-in. of 2-in. pipe. Each riser connects to a radiator through a 2-in. ell and close nipple.

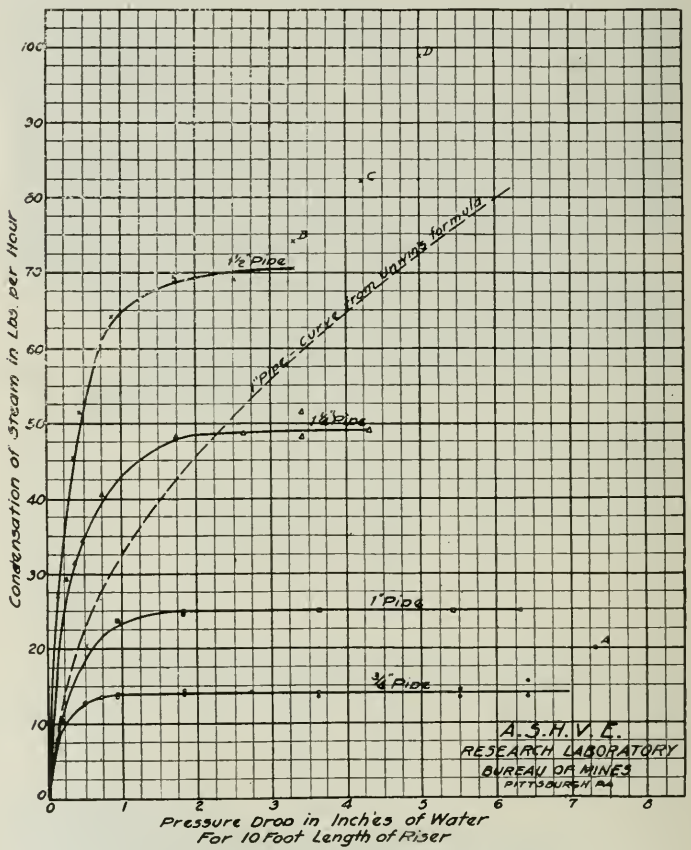

Condensation of Steam in Lbs. per Hour
Pressure Drop in Inches of Water For 10 Foot Length of Riser
1½" Pipe
1" Pipe
¾" Pipe
1" Pipe - Curve from Marks formula
A.S.H.V.E. RESEARCH LABORATORY BUREAU OF MINES PITTSBURGH PA

FIG. 3. CHART SHOWING TEST RESULTS

The ¾-in. and 1-in. risers are connected to radiators of 100 sq. ft. of radiating surface. The 1¼-in. riser is connected to a radiator of 150 sq. ft. of radiating surface, and the 1½-in. riser to a radiator of 200 sq. ft. of radiating surface.

The radiators were each calculated to be large enough to condense all the steam which the risers would carry, with a pressure of a few inches of water in the header. One of the radiators, connected to one of the 1-in. risers, is installed to operate as a two pipe system. All the other radiators are one pipe connected. Manometer tubes for meas-

uring the pressure of the steam are connected to the header and at the top and bottom of each riser. Steam is supplied from the Bureau of Mines power plant. This steam is generated at a pressure of 125 lb. gauge. Before being admitted to the header, the pressure is reduced by means of a fractional valve, a pressure reducing valve and a pressure regulator. Considerable difficulty has been experienced in obtaining steam at the desired constant pressures.

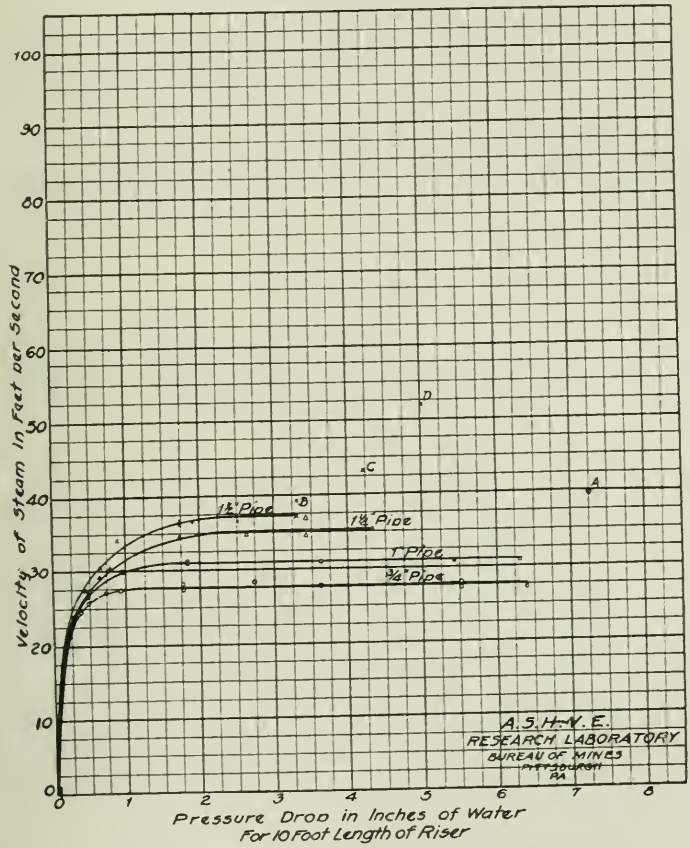


FIG. 4. CHART SHOWING RESULTS OF TESTS ON ONE-PIPE RISERS

In conducting the tests, all the radiators are operated with valves wide open, and with a given pressure in the header. The condensation is weighed at equal intervals over a period of two hours. In the tests so far conducted, the radiators have been operated at atmospheric pressure, the air vents being wide open. However, since the capacity of the radiators is greater than the capacity of the risers, no steam escapes from the air vent. Tests have been conducted with pressures in the header ranging from a few thousandths of an inch up to several inches.

RESULTS

The results of 30 tests on the one pipe risers are plotted in Figs. 3 and 4. In Fig. 3 the total condensation, or the capacity of the pipe for carrying steam with the counterflow of condensed water, is plotted against pressure drop for the ten foot length of pipe. In the same figure is also plotted a curve giving the theoretical capacity of a one inch pipe for dry steam. This curve is plotted from Unwin's formula:

$$w = 87 \sqrt{\frac{pD\,d^5}{(1 + \frac{3.6}{d})\,L}}$$

in which w is the pounds of steam passing per minute,

p the pressure drop between the two ends of the riser in pounds per square inch,

D the density of steam in lb. per cu. ft.,

d internal diameter of the pipe in inches,

L the length of the pipe in feet.

The curve for the 1-in. one pipe riser closely follows the theoretical curve for the same size pipe, carrying dry steam, for pressure drops less than 0.3 in. of water. As the pressure drop increases beyond this point the increase in capacity of the pipe diminishes. For pressure drops greater than 1½ in. of water the capacity of the pipe is practically constant. The other curves show similar characteristics. However, the larger the pipe the less abrupt the departure from the curve for dry steam.

For pressure drops greater than 0.3 in. of water a gurgling sound can be heard in the radiators. There is, however, no noticeable change in the continuity of flow of the condensate until pressure, ranging from about 6 in. of water in the ¾-in. riser to about 4 in. of water in the 1½-in. riser, are reached. Beyond this point, which is not very sharply defined, the return flow of condensate is intermittent. With intermittent flow the capacity of the pipe increases but not consistently. Points *A*, *B*, *C*, and *D* are the results of tests with intermittent flow of condensate. During the test resulting in the point *D* the condensate returned at very regular intervals of 12 minutes. The return of condensate ceased entirely for a period of about 11 minutes and then flowed at full pipe capacity for about one minute. This means that the steam flowed unhindered by returning water for about 11/12 of the time while it was completely shut off for about 1/12 of the time. These conditions would be expected to give a greater flow of steam than where the flow is constantly hindered a lesser amount by the returning water.

In Fig. 4 the velocity of the steam in the one pipe risers is plotted against pressure drop in inches of water. These curves show the same characteristics as those in Fig. 3. The velocity indicated by the pressure at which the capacity of the one pipe riser departs from the theoretical capacity of the same pipe, for dry steam, is the critical velocity for that pipe.

CONCLUSIONS

While sufficient data has not as yet been collected to allow the forming of conclusions which can be accepted as final, certain tentative conclusions may be drawn from the curves. These tentative conclusions are here set forth for the purpose of inviting suggestions, advice and discussion from those interested in the subject. Before these conclusions can be finally accepted as correct they must be verified, by further tests upon the present apparatus and also upon actual installations.

1. There is a maximum capacity of a one pipe riser for the flow of steam, above which any further increase in pressure drop will not increase the flow of steam.

2. The critical velocity of steam and condensate mixtures in one pipe risers, as indicated by the departure of the capacity-pressure drop curve, from the theoretical curve, for dry steam, is approximately 22 ft. per second. This is further verified by the gurgling or surging sounds in the system when this velocity is exceeded.

JOINT DISCUSSION OF PAPERS

on

THE CRITICAL VELOCITY OF STEAM IN ONE-PIPE SYSTEMS

By F. E. Giesecke, Austin, Texas

and

CAPACITIES OF STEAM HEATING MAINS AS AFFECTED BY CRITICAL VELOCITIES OF STEAM AND CONDENSATE MIXTURES

By F. C. Houghten and L. Ebin, Pittsburgh, Pa.

James A. Donnelly: In the preparatory analysis of the problem of critical velocities it became apparent that one of the first and most difficult problems to solve was the matter of obtaining a very exact method of controlling the quantity of steam supplied. This has not been as yet completely worked out, but we believe we are on the right track. The second was to obtain a very refined method of measuring the pressure of the steam.

It was intended that the radiator installed to operate on the two-pipe system should serve as a check on the amount of steam conveyed by the one-pipe risers. In fact, if the amount of steam delivered to the two-pipe riser is carefully calibrated for various pressures, it may be that this will be the most effective means of indicating and reproducing the various pressures desired.

Due to the lack of refinements in the method of supply and indication of pressure, most of the tests have been made at or above the higher limit of critical velocities. This has been, however, no disadvantage, as it has shown in the very early stages of the research, the remarkable flattening out of the curve of amount of steam delivered.

I do not know of anything that I can add to this except to say that the Research Laboratory is undertaking what to my mind is a very important and interesting piece of work. They have a very nice layout of apparatus and a competent staff of observers. I think the work will take quite a long time to complete, and will result in something very useful to all of us.

CHARLES F. EVELETH: We made a number of experiments at our laboratory in Camden in order to determine the proper size of horizontal runouts from radiators to risers. No doubt many engineers and contractors have had trouble in sizing runouts where the radiators are located 10, 12, 15 or 18 ft. away from the risers. With this idea in view we started a series of experiments using commercial steel pipe, having an internal diameter of 1 in., 1¼ in., 1½ in. and 2 in. respectively and approximately 18 ft. long. After making a series of observations we discovered that we did not have the correct conception of critical velocity because of ignorance of what was taking place inside of the pipe. Two glass tubes were accordingly set up, one having an internal diameter of ¾ in. and the other 1⅜ in., to determine visually just what took place when steam at various velocities passed over the condensation in pipes graded against the steam flow. The tubes were placed in a position nearly horizontal and pitching downward slightly toward the supply end. We connected them up to vertical risers arranging the piping in such a way that we could have either an up-feed or a down-feed supply. The volume of steam entering the tubes was controlled by gate valves suitably placed. Provision was also made to vary the pitch or grade of each tube. At all times the pressure of steam in the tubes was slightly above atmosphere. A manometer was arranged to measure the pressure at the entrance to the tube. We found a very interesting set of conditions taking place which we probably would not have discovered if we had confined our observations solely to steel pipes where it was impossible to observe what was going on.

As the tubes were open at the end, the water collecting therein resulted only from the condensation of steam, which was continually taking place.

We started with a very small inclination of the tubes and kept increasing the pitch at the same time watching the action of the steam and water within. When steam was first turned on, a small film of water gathered on the interior of the tube and extended along the bottom perhaps 2 or 3 ft. from the riser. With a relatively low velocity this water would flow back against the current of steam. As we kept increasing the quantity of steam and consequently its velocity, this layer of water kept lengthening out. When it almost reached the far end of the pipe, we stopped any further increase in steam flow and noted the conditions.

As the depth of water in the tube became greater, due to additional condensation of the steam passing over the top of it, slight waves were formed.

A further increase in depth, reduced the effective area of the tube, thus increasing the steam velocity, and the waves kept growing higher and higher. The point was finally reached when the amount of condensation was so great and the steam velocity was so high that the water was ejected out of the end of the tube in a gulp. An elbow was attached to the end of the tube, with a short vertical connection similar to a radiator

runout. Exactly the same result took place, except that the water would become a little deeper in the far end of the tube before it was blown out.

We also noticed a slightly different action taking place in the case of a downfeed steam supply from that of an up-feed supply. With the up-feed riser there was a tendency for the steam entering the horizontal tube to strike against the top of the tube. When there was a down-feed supply the steam was diverted downward toward the bottom of the tube striking the water first. Consequently the same wave formation would take place when the velocity of steam was somewhat less in the down-feed supply than in the up-feed.

From these preliminary tests we concluded that the velocity in a nearly horizontal pipe in which the condensation is to be drained by gravity in the opposite direction to the flow of steam above it, becomes *critical* when it reaches such a rate that any further increase will cause the condensation to be swept upgrade against gravity.

Having defined the critical velocity, we began a second series of experiments using commercial steel pipes and with very low steam pressures, to discover the relationship between the critical velocity and the pitch of the pipes.

Each of the four sizes of pipe, referred to above, was tested at uniform grades of ¼ in., ½ in., 1 in. and 1½ in. in 10 ft. For all velocities below the critical, the condensation forming within the pipes flowed back freely against the current of steam. As soon as the critical velocity was exceeded the condensation was forced out the free ends of the pipes.

It was found that the size of the pipe had substantially no direct relation to the critical velocity and therefore only one size of pipe was tested at a grade of 3 in. in 10 ft.

From these results a curve was plotted using the grade of the pipe in in. as ordinates and the velocity in ft. per min. as abscissae. For a pitch of ½ in. in 10 ft. the critical velocity was found to be 950 ft. per min., for 1 in. in 10 ft. about 1175 ft. per min. and for 3 in. pitch it was 1350 ft. per min. The curve takes the general form of a hyperbola and rapidly approaches the asymptote for values greater than a pitch of 3 in. in 10 ft. The curve also showed a rapidly diminishing increment of critical velocity for pitch greater than 1 in. in 10 ft. and the curves of the entire series of readings indicated that the critical velocity was little if any greater for slopes beyond 3 in. in 10 ft. The action of the condensation within the glass tubes indicates that the critical velocity in a horizontal runout is somewhat lower when taken from a down feed riser than from an up feed riser. This is owing to the high velocity steam at the entrance of the pipe acting upon the condensation which is attempting to flow in the opposite direction.

For practical use in our engineering organization, tables were worked out for sizes of horizontal runouts to radiators for one pipe and two pipe work for various amount of direct radiation.

A description of these tests is contained in our book, *Steam Heating* which has just come from the press and which was previously referred to by Mr. Haynes.

There is one thought I wish to leave with you in connection with the sizes of horizontal runouts. Where surging of condensation is not per-

missible in runouts, the steam must not exceed the critical velocity under the maximum conditions. This occurs during the heating up period when the demand for steam is greatest. Not only does the radiator emit heat but it also requires an additional amount of heat to raise the temperature of the material itself. It is during these times that the hammering and gurgling of water in the runouts cause the most complaint. The flow of steam at the time is most liable to exceed the critical velocity and sweep the condensation up the vertical riser pipe into the inlet valve.

DIRECTOR ANDERSON: The material that has been presented here this morning must strike you gentlemen as very significant. Here are three independent investigators working on the subject of critical velocities and you will note in each case that the velocity is somewhere in the neighborhood of 20 ft. per sec., increasing a little with the larger size pipes.

I want to call your attention to some of the policies in reference to this particular study. This work represents an investigation to meet one very important side of the work of the Society. It is the purpose of the Research Laboratory, so far as we can, to deal with all phases of the work in heating and ventilation. It is not proposed to deal with ventilation entirely nor do we propose to deal with heating entirely. This question of the flow of steam through pipes is one that goes back to the time of Péclet, or pretty nearly 75 years. It seems to me now that we are getting to a point where we will know more about what happens in a pipe when we put steam through it.

To further carry on this work the Carnegie Institute of Technology, through Professor Dibble, has been able to secure a fund of $450.00 for summer work and we have appointed a very pleasant young investigator, L. S. O'Bannon, assistant professor of thermo-dynamics at the University of Kentucky, who will come to Pittsburgh and work with Mr. Houghten and other members of our corps on this very important problem. Similar apparatus to that shown by Mr. Houghten has already been set up in this university and when Mr. O'Bannon returns after his summer in our Research Laboratory we will have two independent but coordinated studies on pipe sizes.

Now what does that mean? We will have this man trained to carry on this work and we send him back to the university, where he will have ten or fifteen competent young engineers, seniors, as research investigators under his direction; so that we will have, gentlemen, by the January meeting, a real check on pipe size data.

With the apparatus that we have and constant contact between the laboratory in Pittsburgh and the one in Kentucky, we think we will be able to determine something final not only with reference to the flow of steam through pipes and specific critical velocities but all those laws of flow of steam from orifices of all sorts and under all conditions.

I want to call your attention to another striking example of cooperation. The Johnson Company has completed, at an expenditure of $25,000, a laboratory, designed for the study of ventilating problems. Through Mr. Ellis of this company this apparatus has been turned over to us to do whatever we please with it. I think such voluntary assistance is extremely gratifying and to find manufacturers so interested in the work of the Society, as to turn over apparatus, that they have devised

for their own use, to the Research Committee, shows the spirit that wins. And gentlemen, in my opinion, we are simply making a start in getting people interested in the work of the Society, and I hope as the days go by we will have more good things to report to you.

Harry M. Hart: I am very glad that this subject has finally become alive, and I think that we have to thank Mr. Donnelly for his persistence in keeping plugging away at it till he got the thing going. At the beginning I was rather skeptical as to the interest that would be displayed in it, but I have got to back water. I know that it is going to be of extreme influence and importance.

Some years ago this Society conducted some experiments at the research laboratory of the American Radiator Company on this same question. In a crude way, sections of glass pipe were inserted in both the horizontal branches, from the main to the riser, and in the vertical riser, so that the action of the condensation of the steam could be observed. It was not as refined as this, but at the same time it was interesting, and I think that those experiments will check up in a way this work that was done. It is interesting to observe the action of this water. As has been stated before, first small waves will start, and as the velocity increases the waves will become higher, as you would naturally expect. We all know that, but what we want to know is just at what point that occurs.

J. M. Robb: There is one point of information in connection with the two papers that I think was not fully brought out; that is the initial pressure of the steam. Mr. Houghten mentioned that in his test the initial pressure was 125 lbs. I did not get from Mr. Eveleth's statement what his initial pressure was. All the tests seemed to show that the pressure of the steam was redueed; and it seems to me you will find quite a difference in your results based on your initial pressure.

Mr. Eveleth: Naturally our experiments were conducted with very low pressure steam, as they were made with a view to practical application in connection with vacuum heating and atmospheric systems of heating. We had no occasion to carry on high pressure tests. Steam was taken at boiler pressure and reduced to a small amount above atmosphere, so that it was necessary to measure the quantity by a manometer tube. I do not happen to carry the exact pressure in mind, but it was very low.

Mr. Robb: The point I wanted to make was this: that the pressure shown by your manometer does not always indicate just what you are doing. Reduced pressure of steam carries some superheat which affects your results; it seems to me important to bear that distinction in mind.

Mr. Eveleth: Our steam was saturated. We did not have thermometers in the glass tube experiment but we had them in the iron pipe experiment and the temperature indicated that the steam was saturated.

Mr. Robb: I do not know whether I fully brought out the point or not, but I noticed in a very large installation where we were working with orifices for a total load of 40,000 sq. ft. of direct steam radiation carried by an 8 in. steam main between 450 and 500 ft. long that a variation in boiler pressure did affect the flow through the orifices although the steam main pressure was constant.

F. C. HOUGHTEN: In some of the first tests made, there was some superheat. This was corrected, however, and in all the tests recorded saturated steam was used. In order to have better control over steam pressure it has been decided to use a small gas fire heating boiler as a source of steam rather than the line from the power plant as heretofore.

The program for the continuation of the investigation includes a study of the flow of steam under higher pressures. That is, you cannot have the air vent open to the atmosphere. Tests will also be made on the glass tubes.

N. L. DANFORTH: This work of standardizing pipe sizes seems most important to me, as nothing makes the heating plant fail more than improper pipe sizes. If we can get pipe sizes standardized so that architects and engineers will use our standards, we contractors will have very little trouble with heating plants, because of improper design.

H. J. MEYER: This discussion has been very interesting and a question that comes to mind is: Why do the Research men make their investigations with the use of such low pressures? It seems to me the pressures that were used, are those never used for single pipe gravity. In fact in our experience we do not expect a single pipe gravity system to work under 1 lb. When exhaust steam is the heating medium, we go to vacuum systems. The pressures used in the tests were 4 in. water pressure which is considerably lower than would be used in single pipe practice. Why not go to practical pressures in such investigations?

DIRECTOR ANDERSON: We are starting at one end of the scale. We will eventually have values throughout the whole range of pressures.

C. V. HAYNES: If we can get these pipe sizes on a standard basis so that engineers from Minnesota, Maine or California will start to use them for different classes of work, we can do exactly the same with other things.

No. 636

THEORY FOR THE FLOW OF CONDENSATION IN RETURN PIPES

By R. V. Frost, Norristown, Pa.
Member

FOR some unknown reason, no definite theory or formula upon the proportions of return pipes, based upon the actual clearly defined facts of the case, has ever been established. Nothing can be found in any of the textbooks, of any consequence, and the authorized tables of leading engineering firms are not based on actual fact. If one attempts to check or compare the various tables, one is struck by the utter lack of agreement between tables that are supposed to cover identically the same conditions.

From such an examination the impression is gained that return pipe proportions are to be regarded as of no consequence and as an unimportant factor in the design of a steam circulating system.

The problems discussed in this paper, first came up in some of the early discussions of the Society's present Committee on Pipe Sizes, when it was endeavoring to obtain a working basis for return pipe proportions.

The most important conditions or factors entering into the proportions of return pipes may be stated as follows:

Wet returns and dry returns are not affected by the same conditions; wet returns having but one element, water, whose flow is affected by head or pressure and by friction; while dry returns have two and sometimes three elements, water, air and steam, all of which react to the impelling forces in different ways.

Considering dry returns first, the following specific conditions must be taken into account.

1. The maximum volume of condensation is encountered when the radiation is first heating up. In considering this, the rate of flow is dependent largely upon the rate of admission of steam to the radiating surface, for when there is an ample supply of steam at the source, the steam supply pipes, if properly proportioned, are able to deliver, at the pressure differential, the full volume of steam that the radiator will hold, at a much faster rate than the radiator trap will permit the exit of air and water. This condition causes the maximum exit of air at the time of maximum condensation. The return pipe must therefore be capable of handling the maximum of condensation and maximum volume of air at the same time.

2. The maximum air volume to be expelled from the radiation is equal to the internal volume of the radiation. By calculation it is found that 1½ in. pipe has the greatest ratio of volume to heating surface of any of the commonly used

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types of radiation, this ratio being 3 cu. ft. per 100 sq. ft. of surface. This quantity is therefore taken as the unit of air flow. The rate of air emission from the radiator is subject to great variation, but as the pipe capacity must be sufficient for extreme cases, 10 minutes is assumed a liberal allowance for the total emission of air from the radiator.

3. The velocity of air flow must be low in order not to require too great a pressure drop. If ¾ oz. pressure drop is allowed the velocity in a 1½ in. pipe will be 20 ft. per second.

This pressure drop would seem to be sufficiently low for the most extreme cases, since the resistance at the exit would be atmospheric or less, and pressure would be required only to overcome friction losses and to give velocity to the air flow. A velocity of 20 ft. per second as a maximum is undoubtedly below the point at which the air would pick up or crowd, and cause to pile, the water of condensation flowing at the bottom of the pipe.

4. The velocity of water flow in a dry return is caused by the slope or gradient of the pipe and is influenced but little, if any, by the pressure drop. Internal friction is the principal retardent to the water flow, and the slope must be in such proportion as to overcome or balance its resistance.

5. If steam is admitted to the return line, as is the case in the typical vapor system, then the pipe capacity must be increased in proportion to the volume of steam admitted. The volume of steam should be treated as increased air volume, and the increase to the condensation allowed for by increasing the radiation condensation factor.

6. The flow of air and water in the return is usually in the same direction, but when the flow of the two elements is in opposite direction to one another, allowance must be made so that the air velocity can be reduced. What this velocity should be, is difficult to determine. As an assumption however, this reduction can be placed at 25 per cent, or a velocity not to exceed 15 ft. per second.

7. Restrictions in the return line such as elbows, probably do not affect the velocity of water flow in the same manner as internal friction, but do affect the carrying capacity of the pipe, through the tendency to pile up the water at the turns, thus reducing the free area for the movement of air.

This last point is one of interesting research, and valuable data affecting return pipe proportions could be obtained from carefully worked out experiments.

FACTORS AFFECTING PROPORTIONS OF RETURN PIPES

From the above stated conditions, the following factors are obtained, affecting return pipe sizes:

1. Volume of condensation, to be stated in square feet of radiation.
2. Velocity of condensation flow, of which the following are sub-factors:
   a. Area of the pipe.
   b. The perimeter of the pipe in contact with the water.
   c. The coefficient of friction for the material of the pipe.
   d. The slope, expressed as a ratio of the height to the length.
3. The volume of air to be discharged, which resolves into three sub-factors:
   a. Time allowance for discharge.
   b. Velocity of air flow.
   c. Free area of pipe for the passage of air.

Observe, that in a dry return where but a portion of the area of the pipe is occupied by water, the frictional resistance is dependent upon the ratio of the cross-sectional area of the water in the pipe to the perimeter of that portion of the pipe in contact with the water.

These two expressions are known in hydraulics as the wet area and the wet perimeter, and the expression for the ratio as the hydraulic mean depth or the hydraulic radius. Thus, when the hydraulic depth or radius is referred to, it is to be understood as the $\frac{\text{wet area.}}{\text{wet perimeter}}$.

There is a formula, well known to civil engineers, but for which mechanical engineers have little use, which takes into account all of the factors affecting the flow of condensation in a dry return. This formula, known as Chezy's formula, was first developed in 1775, and is in general use by hydraulic engineers in the design of open conduits, and in sewerage work in the proportioning of sewerage systems.

It is a formula of such importance, that there has been an unusually large amount of experimental work performed to correct and verify it, particularly the work of Ganguillet and Kutter, the results of whose experiments are recorded in the formula known as Kutter's formula.

Chezy's formula, in the usual form, is expressed as follows:

$$Q = a\,c\sqrt{r\,s}$$

where Q = discharge in cubic feet per second.
a = wet area in square feet.
r = hydraulic mean depth $= \frac{\text{wet area.}}{\text{wet perimeter}}$.
d = actual diameter of pipe in feet.
c = constant from Kutter's formula.
s = slope expressed in ratio of height to length.

In the form in which the formula is used here, Q is reduced to square feet of radiation, or more correctly, the condensation per square foot of radiating surface per hour. The other factors are retained in the form found in the original, that is, in feet, as more convenient than to reduce to inches. The factor r, the $\frac{\text{wet area}}{\text{wet perimeter}}$, takes the form $\frac{\text{segment of a circle}}{\text{arc of the segment}}$. The factor c from Kutter's formula, is derived from the expression

$$C = \frac{41.6 + \frac{0.00281}{s} + \frac{1.811}{n}}{1 + \left(41.6 + \frac{0.00281}{s}\right)\frac{n}{\sqrt{r}}}$$

in which n is the coefficient of roughness for the sides of the pipe, and s and r slope and hydraulic radius as above. The value of n has been determined by experiment and runs from 0.010 for very smooth iron

pipe to 0.012 for cast iron pipe and up to 0.017 for conduits or channels having walls of rubble or granite block.

In working up the table of return pipe sizes from these formulæ, the first question that the Committee found itself compelled to settle, was to determine the proportion of the pipe cross-sectional area that is occupied by the condensation flow. To settle the question, as experimentation was impossible at the time, two tables were worked up, one in which the wet area was taken as ¼ of the total area, and one as ⅛ of the total area of the pipe. Finally 3/16 was decided upon as the proportion more nearly fitting the best practice where thermostatic traps are used, and ¼ as the best proportion for return lines without traps. The tables published in the GUIDE are on this basis, although there is some slight discrepancy to actual calculations, as the values were interpolated from the ¼ and ⅛ tables, instead of being actually calculated out for the accepted proportion of 3/16.

Having determined the proportion for the wet area, the balance of the factors, the wet perimeter, hydraulic radius, and values of the constant from Kutter's formula were matters of calculation only. The diameters for pipe are the actual and not nominal diameters. The slopes are the usual slopes found in heating practice, namely, 1 in. in 10 ft., 1 in 20, and 1 in 30.

Having completed the determination of all factors affecting the flow of condensation, now consider the factors affecting the air flow. As has been stated they are volume, and time, velocity and free area.

TYPICAL EXAMPLE

Taking a specific case: From the tables it is found that 1¼ in. pipe will carry the condensation from 575 sq. ft. of radiation when flowing 3/16 full. The balance of the area of pipe, or free area for the flow of air, equal to $\frac{13}{16} \frac{\pi d^2}{4}$ where $d = 0.115$ ft., has a value of 0.00843 sq. ft. The volume of air from 575 sq. ft. of radiation, at the ratio of 3 cu. ft. per 100 sq. ft. of radiation, equals 17.3 cu. ft. If discharged in 10 minutes, the rate of flow is thus 0.0288 cu. ft. per second, or a velocity through the free area of the pipe of 3.42 ft. per second or 205 ft. per minute. Similarly, the result for a 2 in. pipe, is a velocity of 4.9 ft. per second or 294 ft. per minute, and for a 5 in. pipe, a velocity of 10.5 ft. per second or 630 ft. per minute.

From these figures it is seen that the air velocity is much below the allowable limit of 20 ft. per second and it may be that the allowable limit that has been assumed is much too high. The sizes of returns as worked out on the 3/16 proportion are considered high by some authorities and low by others, but strike a fair average of what may be considered best practice. It is quite possible that the action of the water at the turns, in choking the free air passage, may be the cause for the required low air velocity, a point, as mentioned before, that can only be determined by experimental research.

If it is true, that such low air velocities are necessary, on account of

the choking at the turns, then it is quite evident that by the use of long sweep fittings, this trouble could be largely removed, and as a result the capacities of the return lines somewhat increased.

Turning now to the subject of wet returns, the Chezy formula is also used in hydraulic work for pipes running full, and it would seem that it could be adapted to wet returns as well, but conditions are not quite parallel. The formula is used in hydraulics when the velocity of flow is maintained by the slope, but since the flow in a wet return is actuated by a pressure head, although of necessity very small, the regular formulæ based on Bernoulli's theorem by Unwin and Weisbach and of the same form as the familiar Babcock formula for the flow of steam, is the proper one to use.

A very convenient form for the purpose of computing a wet return is expressed by

$$d = 0.0226\sqrt[5]{r^2 l} \text{ or } r = \sqrt{\frac{\left(\frac{d}{0.0226}\right)^5}{l}} \text{ or } \sqrt{\left(\frac{d}{0.0226}\right)^5 \frac{1}{l}}$$

where d = diameter of pipe in inches.
r = square feet of radiation.
l = length of pipe in feet.

This equation is derived from the formula, found in Brigg's *Steam Heating,*

$$d = 0.5374 \sqrt[5]{\frac{Q^2 r^2 l}{h}}$$

where Q = volume of water condensed per square foot of radiation per minute.
h = head in feet of water required to produce flow.

The equation in the form above presented is based on a head of ½ in. of water.

So far dry and wet returns have been considered only in horizontal position. Return risers obviously can not be proportioned by the aid of either the Chezy or Unwin formula, but it is doubtful whether the flow of condensation in the riser need be taken into account, allowance being made for the passage of air only. Of course if there are horizontal steps in the return, then the horizontal formula must be applied, but if proper allowance is made for the volume of air to be moved in the vertical riser, then there is ample capacity for the movement of condensation.

Most of the ideas presented in this paper are new in the heating field and the author realizes that they offer plenty of opportunity for discussion and research in order to prove out whether all the assertions made, are sound. Therefore all the constructive criticism that may be offered will be most gladly welcomed.

Data sheets Tables 1 and 2, show the method of calculation for both dry and wet returns.

TABLE 1. DATA FOR DRY RETURN PIPES. CONDENSATION FILLING $\frac{3}{16}$ OF PIPE AREA. GRADE 1 IN. IN 20 FT.

$$s = \frac{1}{240} \qquad \sqrt{s} = 0.0645$$

Pipe size	$a = \frac{3}{16}\frac{\pi d^2}{4} =$ 0.1475 d^2	$c = \frac{207.275}{1 + \frac{0.465}{\sqrt{r}}}$	$\sqrt{r} = \sqrt{\frac{0.1475 d^2}{1.13 d}} =$ 0.361$\sqrt{d}$	$d =$ actual diam. of pipe in feet	$Q =$ cu. ft. per sec. $= a\,c\sqrt{r\,s}$	Sq. ft. rad. $= 865000\,Q$	$v =$ ft. per sec. $= c\sqrt{r\,s}$	Velocity ft. per min.	Free area $= \frac{13}{16}\frac{\pi d^2}{4} =$ 0.638d^2	Total air volume at 3 cu. ft. per 100 sq. ft.	Air C.F.M. at 10 min. for total discharge	Velocity ft. per min. through free area
½	0.000398	31.2	0.0823	0.052	0.000066	57	0.166	10	0.001725	1.71	0.171	100
¾	0.000694	35.0	0.0945	0.0688	0.000148	128	0.213	13	0.00300	3.84	0.384	128
1	0.00112	38.7	0.1068	0.0873	0.000299	258	0.267	16	0.00485	7.75	0.775	160
1¼	0.00195	43.2	0.1222	0.115	0.000665	575	0.341	21	0.00843	17.25	1.725	205
1½	0.00266	45.0	0.1320	0.134	0.00102	885	0.384	23	0.0115	26.60	2.66	231
2	0.00435	50.5	0.1500	0.172	0.00213	1840	0.490	29	0.0188	55.25	5.525	294
2½	0.0062	53.9	0.1635	0.205	0.00353	3050	0.570	34	0.0268	91.50	9.150	340
3	0.0096	58.3	0.1820	0.255	0.00658	5700	0.685	41	0.0415	171.0	17.10	415
3½	0.0129	61.5	0.1960	0.296	0.01003	8700	0.780	47	0.0558	261.0	26.10	460
4	0.0165	64.5	0.2090	0.335	0.01435	12400	0.870	52	0.0715	372.0	37.20	520
5	0.0260	69.4	0.2340	0.420	0.02725	23600	1.05	63	0.1125	708.0	70.80	630

TABLE 2. DATA FOR WET RETURN PIPES, 100 FT. LENGTH

$$r = \sqrt{\left(\frac{d}{0.0226}\right)^{\frac{5}{2}} \frac{1}{l}}$$

r = sq. ft. radiation
d = actual diam. in inches

$$\sqrt{\frac{1}{l}} = \sqrt{\frac{1}{100}} = \frac{1}{10}$$

Pipe size	d Inches	$\left(\frac{d}{0.0226}\right)^{\frac{5}{2}}$ log	r Sq. ft. rad. ¼ lb. per ft.
½	0.625	3.6044290	400
¾	0.825	3.9059	800
1	1.05	4.1677	1500
1¼	1.38	4.4642	2900
1½	1.61	4.6315	4300
2	2.065	4.9019	8000
2½	2.46	5.0919	12400
3	3.06	5.3287	21300
3½	3.55	5.4900	30900
4	4.02	5.6250	42200
5	5.04	5.8708052	74200

DISCUSSION

C. V. Haynes: In preparing this paper Mr. Frost communicated with nearly 50 concerns in the United States who have tables of sizes for steam mains, branches and returns, and they show variations of 600 per cent. The table prepared for the A. S. H. & V. E. Guide checks up very closely with the pipe sizes used by Warren Webster & Co., James A. Donnelly and other concerns engaged in practical work.

Checking up Mr. Frost's paper and comparing it under these conditions with various other tables from the best authorities we find that these tables show approximately ¼ in. drop per 100 ft. of straight run and check up all the way through.

James A. Donnelly: I am very glad to see the paper on return pipe sizes because there have only been a very few presentations of this subject, and as they were some time ago, it will undoubtedly be profitable to review the subject again at this time.

For many years, the only figures given for return mains were comparative ones; for a certain size steam main there would be a corresponding size of return main given, without regard to the amount of water, air or steam it might be called upon to carry. At that time, also, there was little if any distinction made between the sizes of a wet or dry return.

The title of this paper should indicate its contents as applying only to vapor systems, as the author does not mention either vacuum return line, ordinary two-pipe gravity systems, or the drip lines from one-pipe systems. Neither does he discuss the sizes of returns where float or mechanical valves, or ordinary or special forms of check valves are used on the return.

The preferable size of return mains depends very much upon the point of view of the person who selects the "actual, clearly defined facts," as the author terms them. The promoter of special apparatus may assume that his equipment always functions 100 per cent. The operating engineer, however, may prefer pipe sizes that will be large enough to give him good operation if the automatic devices fail, partially or entirely, in their function, or if he wishes to remove them for cleaning or inspection and meanwhile blow the system out.

It would certainly be interesting to establish safe return pipe sizes for both conditions and give comparisons of first cost. Then the consulting engineer and owner would have a wider latitude of choice.

In my opinion, the division of dry returns for vapor systems into two classes—one with and one without the use of thermostatic traps—should be made upon the ground that the one not using traps is certain to have steam in the returns in cases of excessive pressure. This is also true of plants equipped with thermostatic traps, due to the re-evaporation in the return lines.

The assumption, or perhaps rough guess would be the better term, that dry returns run with any definite proportion of their area occupied by water, has a very vague basis in fact, if it has any, for even one particular pitch or condition of operation. However, in all formulae or standards of pipe sizes for heating apparatus, we must assume certain things. We

start off with an assumption 1 oz. to 100 ft. drop represents good practice, in steam main sizes. Then we must make some assumption as to unit pressure drop for steam and water carried in the returns, or for water only in the case of wet returns.

The author has used, he states, a head of ½ in. of water while in all the work which I have done, I have assumed the same head as for the steam main; namely, 1 oz. per 100 feet.

Mr. Frost says in the early part of his paper that the maximum volume of condensation is encountered when the radiation is first heating up, and from the way in which he continues to treat the subject, I rather expected that his table of pipe sizes would perhaps be based on handling the initial condensation. However, upon analyzing the capacity of a 1 in. pipe as given in Table 1, by the factor of 0.000299 cu. ft. of water per second, and 258 sq. ft. of radiation, I find that this is based upon one-quarter (0.25) pound of condensation per square foot.

I note that the wet return, as stated, is based on the same amount. I think it is decidedly better to use 0.3 lb. as this allows for the condensation of the average piping system, in addition to the radiator condensation.

One of the easiest ways of checking up tabulations such as these, is to reduce the capacities of several sizes of pipe, to comparative figures. This is readily done by dividing the capacity of the sizes from 1-in. to 4-in., for dry returns, by the capacity given for the 1-in. size, 258 sq. ft. of radiation. This would make the comparative carrying capacity of a 1-in. pipe to a 4-in. pipe as 1 to 49. Applying the same method to the capacity of the wet return pipes, the proportion between a 1-in. and 4-in. pipe in the latter case is only as 1 to 28. According to the Weisbach formula, the proportion of 1 to 4 in. pipe is as the 2½ power of the diameter, or 1 to 32. Unwin's modification of this increases this proportion to 1 to 49. Mr. Frost states that the formula for wet returns is of the same form as the Babcock and Unwin, but if this is the case, perhaps an error has been made in applying it.

The following table shows the former compilations of the Committee on Steam and Return Pipe Sizes, in comparison with the present ones by Mr. Frost, as printed in the A. S. H. & V. E. GUIDE.

Dia. pipe inches	Two-pipe Gravity Dry Return	Former Vapor Dry Return	Vapor Dry Return (Frost)	Former Wet Return	Wet Return (Frost)
1″	75	500	170- 630	1600	1500
2″	500	4000	1200- 4500	12000	8000
3″	2800	12000	3700-13500	36000	21300

This is a subject upon which I have spent much time and thought, and I am very glad indeed to see that it is going to be given some consideration by others. It is possible to obtain considerable further information on this subject, which will be of practical value. I hope other members will follow the good example of Mr. Frost and present their views.

L. L. BARRETT (written): In the treatment of wet returns it is thought that correction should be made of the author's implication that the Chezy formula can be used only when the velocity of flow is maintained by the

slope and not when it is maintained by a pressure head. In the Chezy formula the hydraulic gradient is equal to h/l, where h is the loss of head due to friction in a pipe of length l. In the application of this formula it is immaterial whether the head necessary to overcome the friction is secured by the slope of the pipe line, or by the initial pressure, or by both. The formula is therefore entirely applicable to a pipe running under pressure. In this case the hydraulic gradient is not, of course, equal to the actual gradient on which the pipe is laid.

It is also interesting to note that the Briggs formula given is nothing more than a rearrangement of the Chezy formula after placing C in the Chezy formula equal to 100. In rearranging the Chezy formula to obtain the Briggs formula it is only necessary to place $s = h/l$, take Q in cubic feet per minute and d in inches, and solve for d, remembering that the hydraulic radius is equal to one-fourth the diameter when the pipe is running full. The Briggs formula is a convenient form for use in the heating field and is correct when conditions are such that a value of 100 can be used for C in the Chezy formula. It must be remembered, however, that C has varying values depending upon the pipe size and velocity used.

Heating engineers should have no hesitancy in using the Chezy formula for the proportioning of wet returns and should so use it whenever there is need for the greater accuracy that comes with the use of a coefficient C which can be made to fit the conditions of the particular case. This degree of accuracy is not necessary on ordinary work but would be indicated in cases such as the determination of return pipe sizes in an underground district heating system where there are horizontal runs of considerable length.

R. V. Frost (written): The author accepts Mr. Barrett's corrections regarding the use of the Chezy formula for wet returns and also for the derivation of Brigg's formula.

Referring to the tables, it will be noted, in the column of velocities in feet per minute for flow of water, that for the given conditions, this velocity is 21 ft. per minute through 1¼ in. pipe, while the velocity for flow of air through the free area of the same pipe is given as 205—a ratio of air flow to condensation flow of 10 to 1.

This condition is not in agreement with the theory, commonly held, that the condensate in a dry return is a saturated solution of air and water and as a consequence the two elements flow at the same velocity.

Since the discussion in this paper is based on the assumption that air and water do not flow at the same velocity, this idea will remain a point for argument until the truth or falsity of it is definitely established.

No. 637

TEMPERATURE CONTROL BY FRACTIONAL DISTRIBUTION OF STEAM IN LARGE ONE- AND TWO-PIPE SYSTEMS

By James A. Donnelly, New York, N. Y.
Member

This article on temperature control by fractional distribution will supplement the data on this subject previously contributed by the author and Alphonse A. Adler under the title Fractional Distribution in Steam Heating Systems, published in the Transactions *1921, p. 297.*

In the present instance the use of this method of control has been applied to one-pipe steam systems, where in the former it was considered only in its application to two-pipe installations.

THE records, recently compiled by members of the *National District Heating Association,* of the amounts of steam actually used in the heating of buildings at various outside temperatures, indicate the great possibility of efficiency and economy by correct control. Given the figures showing the lowest possible rate of use of steam for any particular outside weather conditions, and means for checking up the actual rate of supply during these varying conditions, it is instantly apparent that there should be a possibility of regulating the latter in correct proportion to the former.

Damper control at the boiler, or on street steam systems at the source of steam supply, operated automatically by the amount of returning condensation in connection with restricted orifices at the radiators, has been found to easily maintain very effective temperature regulation. This method, regulating as it does the rate of steam supply to correctly meet the demand, would seem to need no further argument than the mere statement of facts, when compared with the present methods of unlimited steam supply for all conditions of operation.

There are several methods now in use, which aim at economy. In small buildings, temperature regulation from one typical point, hand throttling of the main supply valve, or of the individual radiator valves in air-return vapor systems, are recommended, but are rather limited in application. In large buildings, a complete system of temperature regu-

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lation is applied individually to all radiators, but there are several serious objections to automatic temperature regulation of this kind. It is expensive to install and maintain in perfect operative condition, the source of very severe complaints when in poor repair, and not at all economical if outside windows are opened for fresh air or ventilation. In fact, the amount of heat that may be wasted, either wilfully or ignorantly, is practically unlimited with any of the present methods of temperature control. Because of partially opened windows, when the outside tem-

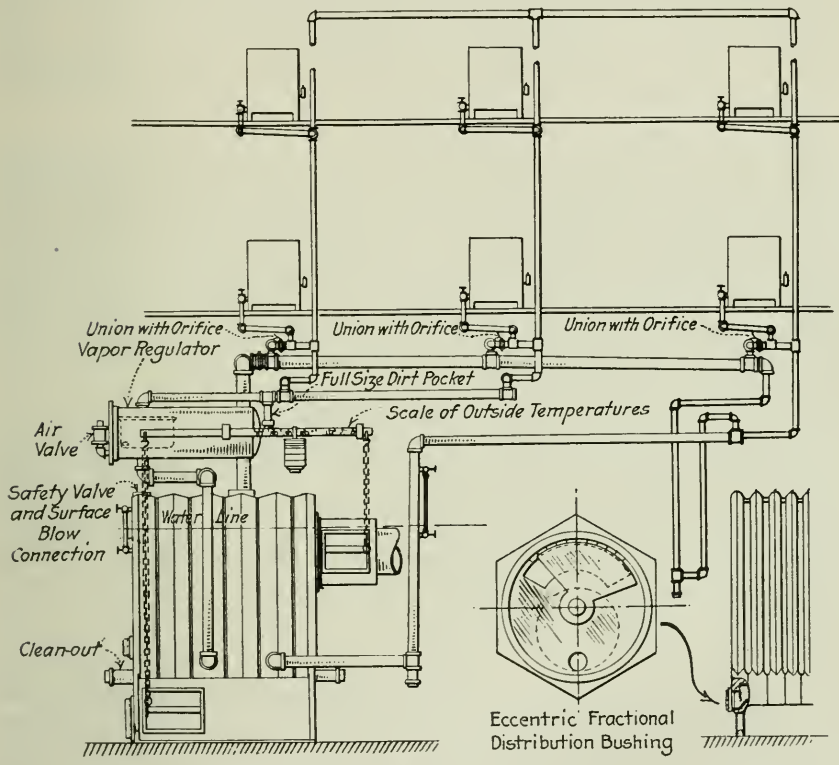

FIG. 1. LARGE ONE PIPE SYSTEM WITH FRACTIONAL STEAM DISTRIBUTION ATTACHMENTS

FIG. 2. RADIATOR WITH FRACTIONAL DISTRIBUTION BUSHING FOR CONTROL OF STEAM

perature is at 35 deg., it is a frequent occurrence to have the amount of steam used to heat a building increase to double, and sometimes three times the amount it should be for that temperature condition.

In small heating systems, where the amount of steam generated by the boiler is automatically controlled in proportion to outside temperature demand, it is comparatively easy to obtain a sufficiently uniform distribution of steam by restricting orifices located at the inlet of each radiator. In larger systems, and perhaps more especially in one-pipe systems, where the drop in pressure becomes a considerable amount, it is not possible to make the pressure drop through the orifice at the radiator inlet sufficient

to compensate for the line loss. In two-pipe systems, undesirably high velocities are necessary through the inlet valves, often causing unusual noises and perhaps making adjustment difficult.

For the length of steam main necessary for the average house, very little special adjustment of distribution orifices at radiator inlets is required, either in one or two-pipe systems. An orifice having direct ratio to the amount of radiation supplied is usually sufficient and additional modification for distance or varying pipe line drop is not required. In large plants, however, where the drop in pressure in the main becomes of considerable amount, it is very necessary to make the allowance. This has often been very successfully accomplished in two-pipe systems, but it is only recently that a method has been developed for accomplishing it in one-pipe systems.

In Fig. 1 a typical one-pipe system of the larger size is illustrated. As the steam mains are laid out on the standard unit basis of 1 oz. drop per 100 ft., the pressure drop to all distances may be pre-determined and the necessary area of orifice for use in the steam riser connection for all varying amounts of radiation and distances are readily tabulated. All first floor radiator supplies are then taken from the riser connection between the orifice and the base of the riser and all risers dripped. This results in a two-pipe system of piping so far as the distribution mains are concerned, and makes correct fractional distribution possible to the base of risers for all loads.

This method of equalizing the distribution of steam to the various risers is recommended for both one and two-pipe systems. The steam main may be proportioned for any drop in pressure allowable by the particular conditions and separately dripped to the return. The risers may then be laid out with due regard to critical velocities and it is also recommended that they be connected together at the top. It is possible to use somewhat smaller pipe sizes where this is done and it also insures a smooth and noiseless starting up of the system.

From observations made recently on the fractional distribution of steam, it has become clearly apparent that the buoyant effect of the steam or its tendency to rise in a vertical pipe makes the distribution through vertical piping a much simpler matter than through horizontal piping. The difference in weight between the column of steam and air is such that it very nearly offsets the friction of flow for the average fractional load.

In order to complete the correct fractional distribution to each radiator, for one-pipe systems, the fractional distribution bushing is provided as shown in Fig. 2. It is used in a drop hub steam type radiator, so that the water seal may be confined to the hub and not extend through the base of the radiator. The adjustable port is set with an area of about 0.0075 sq. in. per sq. ft. of radiation. This gives a drop in pressure through it of about 0.22 oz. or $\frac{3}{8}$ in. of water column. The bushing is of the eccentric type so that the water may drain out through the lower port in the diaphragm.

No. 638

RECENT DEVELOPMENTS IN WARM AIR FURNACE HEATING

By F. R. Still, Detroit, Mich.
Member

SOME years ago the furnace manufacturers formed an organization known as the *National Warm Air Heating & Ventilating Association*. In 1918 this association entered into an agreement with the University of Illinois to have the latter conduct experiments on warm air furnaces. The University has rendered three reports in bulletin form known as No. 112, 117 and 120. The last report, which was presented to the meeting of the association at Cleveland, April 19-20, 1922, has not yet been published. The work so far completed at the University has made available a wealth of valuable data which never before was obtainable. With this data in hand it is now possible to calculate the size and performance of a furnace with about the same degree of accuracy as the size and performance of a steam or hot water boiler can be determined. The unfortunate part, however, is that very few of those who have access to the reports know how to make practical use of the data given in them and these reports have been confined almost entirely to the members of the furnace manufacturers association, who have made no effort to see that they are distributed among their agents and dealers so that they could take advantage of their practical value should they be so inclined.

Like our own Society, this association has a Warm Air Furnace Code Committee which has been struggling long and arduously in an endeavor to reconcile all the elements so as to get them to agree on a standard code. This committee submitted a report at the last meeting and a code was adopted. It is not perfect any more than was the first report on the boiler code, which was submitted to our Society, but it is a step in the right direction, is founded on the right principles and can be modified and amended at subsequent meetings. Beyond a doubt it will eventually be perfected when all the members of the association become more familiar with it and understand the benefits to be derived by having a standard which is followed by everybody.

The formula adopted in the code for determining the size of the leaders, or pipes from the furnace to the various flues and registers was devised by P. J. Dougherty, engineer of the International Heater Co., a member of our Society and also on the Research Committee of the

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

association referred to previously. It is very simple and is based on experimental engineering data taken from the report issued by the University of Illinois. It has been checked by application to certain test problems covering quite a wide range of capacities and it has been found accurate, conservative and reliable.

The rule proposed by Mr. Dougherty is based on the following:

Outside temperature zero deg. fahr.
Inside temperature 70 deg. fahr.

Heat loss per hour per sq. ft. of exposed wall surface of ordinary frame construction ...	25 B.t.u. (0.36 x 70 = 25.2 B.t.u.)
Heat loss per sq. ft. of glass per hour	83 B.t.u. (1.18 x 70 = 82.6 B.t.u.)
Heat loss per cu. ft. air space in a room, allowing one air change per hour	1.25 B.t.u. (0.075 x 0.238 x 70 = 1.25 B.t.u.)

If 1000 is divided by each of the factors, the result will give the square feet of wall surface, the square feet of glass surface, and the cubic feet of space, respectively, to equal 1000 B.t.u. loss per hour.

For example, $\frac{1000}{25} = 40$ sq. ft. of wall surface per M. B.t.u.

$\frac{1000}{83} = 12$ sq. ft. of glass surface per M. B.t.u.

$\frac{1000}{1.25} = 800$ cu. ft. of space in a room per M. B.t.u.

Suppose a room 10 x 12 ft. and 9 ft. high, has 2 windows, each about 16 sq. ft. area. If two of the 10 ft. walls have outside exposure and only one of the 12 ft. walls is so exposed, then the wall exposure will be 10 x 9 x 2 ft., or 180 sq. ft., plus 12 x 9 ft., or 108 sq. ft. which equals 288 sq. ft. Deducting 32 sq. ft. for the 2 windows, leaves a net wall area of 256 sq. ft. A room 10 x 12 x 9 ft. contains 1080 cu. ft. of space.

By dividing each of these quantities by their corresponding factors, as shown above, the result will be the heat units required in thousands; all three should be added together to get the total heat required for the room. This can be stated more clearly as follows:

$$\frac{256}{40} + \frac{32}{12} + \frac{1080}{800} = 10.41 \text{ or } 10{,}410 \text{ B.t.u.}$$

In order to determine what size the leader or pipe should be from the furnace to this room, consult Fig. 1, which is a duplication of the chart given on page 24 of Bulletin No. 120, published by the University of Illinois, and it is found that at a temperature of about 195 deg. there was obtained 125 heat units per sq. in. of leader per hour to the first

story. Hence dividing 1,000 by 125 will give us 8 sq. in. per 1,000 heat units.

Therefore 10.4 x 8 = 83.28 sq. in. area of pipes or leaders to the room if it is on the first floor. This is about the area of a pipe 10 in. in diameter.

If the same temperature of 195 deg. is assumed for determining the second and third story pipes or leaders, the area of them should be respectively 5 sq. in. to the second story and 4 sq. in. to the third story, per thousand heat units exposure. Thus, for a similar room on the second floor, the area of the pipes would be 52 sq. in. or an 8 in. diameter

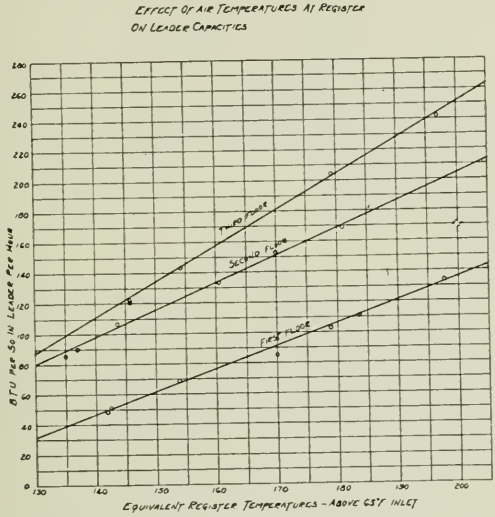
EFFECT OF AIR TEMPERATURES AT REGISTER
ON LEADER CAPACITIES
B.T.U. PER SQ.IN. LEADER PER HOUR
THIRD FLOOR
SECOND FLOOR
FIRST FLOOR
EQUIVALENT REGISTER TEMPERATURES - ABOVE 65° INLET

FIG. 1. CHART FOR DETERMINING LEADER PIPE SIZES

pipe to the second story, and 41.64 sq. in. or a 7 in. diameter pipe to the third story.

It was decided by the Committee to allow 6 sq. in. to the second story, and 5 sq. in. to the third story, which is equivalent to a temperature of about 176 deg. Thus, the pipe to the second story would be 9 in. diameter, and to the third story 8 in. diameter. Undoubtedly, the code will be amended so as to provide for a lower temperature to the first story, the same as has been provided for the second and third. This will mean allowing 9 sq. in. per thousand heat units, or 93.6 sq. in. in the leader pipe, or 11 in. diameter instead of 10 in. as the present code provides in the foregoing example.

Reducing the allowable temperature from 195 deg. to 176 deg. evidently increases the area of the leaders to the first story more than 20 per cent above what is now the average practice, but it will avoid the necessity for maintaining such high temperatures to get sufficient heat, will be conducive to economy, and will avoid so many unsatisfactory plants.

In everyday practice when calculating the size of the leaders to the various rooms in the building, one would proceed about as follows:

Wall 10 ft. + 10 ft. + 12 ft. x 9 ft. = 288 sq. ft.
Glass 16 sq. ft. x 2 = 32 sq. ft. = G.

Net wall surface = 256 sq. ft. = W.
10 ft. x 12 ft. x 9 ft. = 1080 cu. ft. space = C.

C = 1080 ÷ 800 = 1.35 M. B.t.u.
W = 256 ÷ 40 = 6.40 M. B.t.u.
G = 32 ÷ 12 = 2.66 M. B.t.u.

(Heat units in thousands) 10.41 x 8 = 83.28 sq. in. = 10 in. dia. 1st floor
10.41 x 6 = 62.46 sq. in. = 9 in. dia. 2nd floor
10.41 x 5 = 52.05 sq. in. = 8 in. dia. 3rd floor

Should it be the reader's opinion that an air change 1½ times per hr. should be provided for, instead of 1 air change per hour, then substitute 600 in place of the factor 800; or if it is thought that 2 air changes per hour should be allowed because of unusual exposure to high winds, or the frequent opening of doors, or because of poor construction, then use 400 in place of 800.

It is seldom that a contractor will undertake to connect a leader from a furnace to a sun-room and agree to heat it to the same comfortable temperature as any other room in the house. This is probably due to two reasons, one being that the sun-room usually is at a very considerable distance from the furnace and the other reason is due to the unsatisfactory results he has obtained in his efforts to heat such a room. This latter has likely been caused by the fact that he has never realized that fully one-third more capacity would be required in the leader pipes than would be required for a similar room of usual construction.

As an example: consider a room of the same size as in the previous case, but one that has 160 sq. ft. of glass instead of 32 sq. ft. as before, and owing to this large glass exposure 2 air changes per hour will have to be provided for; the size of the leaders would then be as follows:

Total wall surface = 288 sq. ft.
Total glass surface = 160 sq. ft.

Net wall surface = 128 sq. ft.
Cu. ft. space = 1080

1080 ÷ 400 = 2.70
128 ÷ 40 = 3.20
160 ÷ 12 = 13.33

19.23 x 8 = 153.84 sq. in. = 14 in. dia. pipe
19.23 x 6 = 115.38 sq. in. = 12 in. dia. pipe
19.23 x 5 = 96.15 sq. in. = 11 in. dia. pipe

It will be noted that the area of the leaders is almost double the size required in the previous instance.

NEED FOR INCREASED WALL STACK SIZES

One phase of the furnace business, which has not yet been reported on by the University, but which has a very important bearing on the satisfactory operation of furnaces, is the size of the flues in the walls for conveying the heat to the upper stories. The Code Committee made practically no recommendations on this point at all.

The average wall in houses heated by furnaces is framed with studs known commercially as 2 x 4's; they are more often 1⅞ in. x 3¾ in. and are usually spaced 16 in. centers. Thus the maximum space for a sheet metal flue is 3¾ in. x 14½ in.; therefore, it is hardly practical to figure on a flue being more than 3½ in. deep x 13¾ in. wide and experience shows the standard maximum width to be about 13 in.

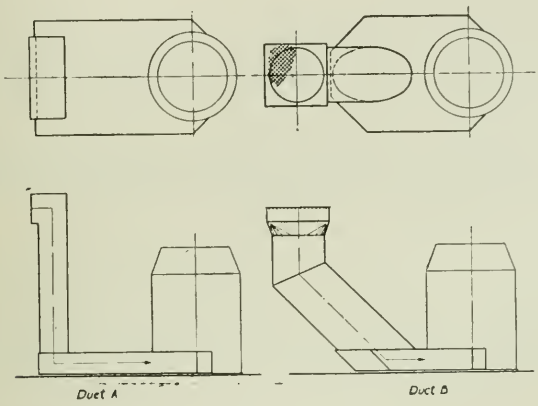

FIG. 2. TWO TYPES OF RECIRCULATING DUCTS CHANGE FROM "A" TO "B' WILL SHOW SURPRISING RESULTS

A round pipe offering the same friction for the same capacity as a rectangular duct of the above dimensions, would be 7⅛ in. in diameter or it would have an equivalent area of 40 sq. in., whereas it has an actual area of 48.25 sq. in. Many are the instances where an 8 in. or a 9 in. leader is carried from the furnace to a riser of no greater dimensions than above mentioned, which only has the carrying capacity of a 7 in. round pipe. It is no wonder that it is so often difficult to heat rooms in the upper stories of buildings under these circumstances.

The way to overcome this, is to insist that all walls carrying warm air flues are to be built of studs not less than 2 in. x 6 in. spaced 20 in. centers. If 6 in. studs are used in such partitions or walls, even though the studs are set 16 in. centers, which is unnecessary, then a flue measuring 5½ x 13 in. could be built which is equivalent to 8¼ in. diameter pipe. When the studs are spaced 20 in. centers then a flue 5½ x 17 in. can be built in, which is equivalent to a round pipe 10 in. diameter in carrying capacity.

In Bulletin No. 120, issued by the University of Illinois, is a chart on page 21 which is herein reproduced in Fig. 3. By means of this chart the size of the furnace can be readily determined, also the amount

of the fuel to be burned, the draft necessary to burn that amount of fuel, the maximum temperature attainable, the heating capacity and the overall efficiency.

As previously stated, the code as adopted, makes it necessary to carry a temperature of 195 deg. in order that the required amount of heat is supplied to first story rooms, when the leaders are determined by allowing only 8 sq. in. of area to each 1,000 heat units of exposure in the building.

Comparing the effect this high temperature has, as compared with what would be the result if only 176 deg. had to be attained, it is found that the rate of combustion must be 6.75 lb. of coal per sq. ft. of grate surface per hour. This will necessitate a draft of 0.085 in. The heating capacity of the furnace is 140,000 B.t.u. per hour, and the overall efficiency will be 56 per cent.

At 176 deg., the rate of combustion will be only 5.5 lb. per sq. ft. of grate surface per hour. The draft need only be 0.07 in.; the heating capacity is 120,000 B.t.u. per hour and the overall efficiency will be increased to 60 per cent.

Assume a house has a total exposure of 120,000 B.t.u. and compare the operating results at the two temperatures, first determining the requirements for 195 deg.

$$\frac{120{,}000}{0.56 \times 12{,}000 \times 6.75} = 2.66 \text{ sq. ft. of grate surface, or } 22.10 \text{ in. diameter.}$$

Heating 1 lb. of air from 65 deg. to 195 deg., or through 130 deg. will require 31 heat units; therefore, 120,000 heat units exposure will require 3,870 lb. of air per hour or about 860 cu. ft. per minute at 70 deg. temperature.

For 176 deg. temperature the following results are obtained:

$$\frac{120{,}000}{0.60 \times 12{,}000 \times 5.5} = 3.03 \text{ sq. ft. of grate surface, or } 23.6 \text{ in. diameter.}$$

Heating one pound of air from 65 deg. to 176 deg., or through 111 deg., will require 26.4 heat units; thus, 4,550 lb. of air per hour must be heated, or about 1,000 cu. ft. of air per minute.

In the first case the coal burned will amount to 17.86 lb. per hour; in the second, 16 2/3 lb. per hour. From this it will be noted that the rate of combustion is about 23 per cent higher to get the high temperature, and the coal consumed to give off the same amount of heat is 7 per cent more.

The volume of air at high temperature is less by about 14 per cent, hence smaller pipes are required. The grate area is about 7 per cent more for the lower temperature; in other words, the furnace having a larger grate surface would proportionately have more heating surface; thus, the same transmission could be effected at lower temperatures, by the circulation of more air, requiring less coal and still have a reserve capacity for quick heating by running the temperature up higher, which would not be possible with the smaller outfit, which is already working at about its maximum capacity.

At Urbana it was found that by making the return air duct to the furnace the same size, or even larger than the combined area of all the leaders, and at the same time avoiding abrupt or right angle turns in this duct, that the capacity of the furnace can be increased as much as 50 per cent. It was further demonstrated that the furnace is affected much more by any restriction to the flow of air in the recirculating duct, than by a similar restriction in the leaders from the furnace to the registers.

One frequently finds installations where there are two or more recirculating ducts leading back to one furnace. This generally fails to give as good results as one duct. It is perfectly obvious why this should be so. For example, one duct 32 in. dia., has about 8 per cent less

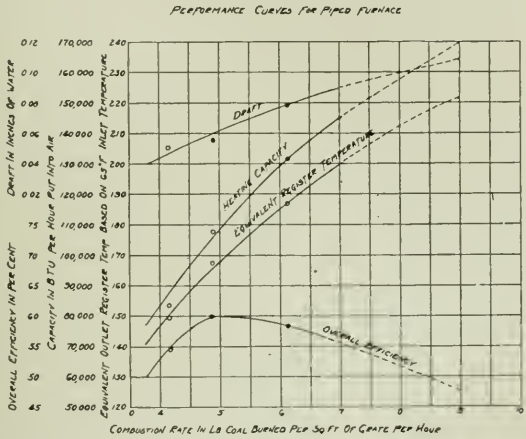
PERFORMANCE CURVES FOR PIPED FURNACE
DRAFT IN INCHES OF WATER
CAPACITY IN B.T.U. PER HOUR PUT INTO AIR
EQUIVALENT OUTLET REGISTER TEMP. BASED ON 65° F. INLET TEMPERATURE
OVERALL EFFICIENCY IN PER CENT
DRAFT
HEATING CAPACITY
EQUIVALENT REGISTER TEMPERATURE
OVERALL EFFICIENCY
COMBUSTION RATE IN LB. COAL BURNED PER SQ. FT. OF GRATE PER HOUR

FIG. 3. CHART WILL GIVE FURNACE SIZE, FUEL AND DRAFT NEEDED, TEMPERATURE ATTAINABLE, HEATING CAPACITY AND OVERALL EFFICIENCY

area than would two 23 in. ducts, yet the total circumference of the two 23 in. pipes is 44 per cent more than the one 32 in. pipe; hence, for a similar length the two pipes present 44 per cent more frictional surface.

One frequently settles on the area to have and because it may be easier to install, uses a rectangular duct instead of a round one. At Urbana there was an opening 17 x 47½ in. behind a register face, with a right angle elbow to a vertical riser of the same dimensions, the lower end of which connected to a horizontal duct 12 x 67 in., the opposite end of which was attached to the base of the furnace. This is indicated in Fig. 2 as shown by *A* at the left. The modified recirculating duct is indicated by *B* on the right.

Both of the above ducts are exactly the area of a 32 in. round pipe, but the drop leg is only equal to a pipe 30.6 in. in diameter as a conductor of air, owing to the shape of it, and the horizontal leg is only equal to a pipe 29.2 in. diameter for the same reason. In other words, the latter can only convey 83 per cent of the air that a 32 in. round pipe will convey because of the additional friction, when both have the same pressure head. Besides this, the right angle elbow alone presents an

amount of resistance equal to 12 ft. of straight pipe 17 x 47½ in. in size, and the right angle at the base will add as much more. In view of all this, is it surprising that the capacity of a furnace can be increased 50 per cent by a few changes in the recirculating duct? Hardly anyone would knowingly set out to install a recirculating duct measuring 17 x 47½ in. that would be nearly 50 ft. in length with a standard register face on the opposite end of it. He would at least feel very uncertain of the results he would get, yet that is exactly what the resistance amounts to in the installation as it was first put in Urbana and it was in no

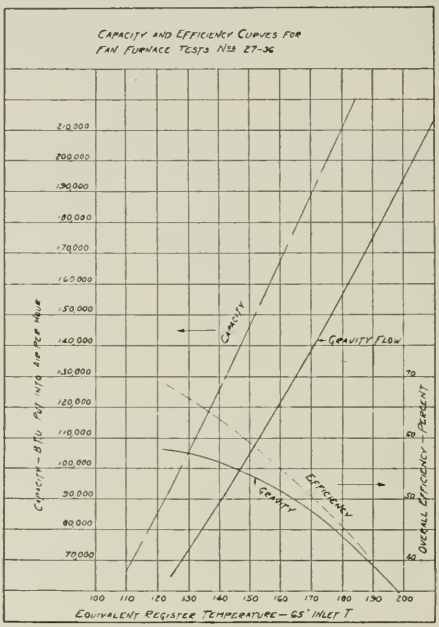

FIG. 4. CHART SHOWS COMPARATIVE RESULTS OF FURNACE TESTED WITH FORCED AND GRAVITY CIRCULATION

way unusual, as it is being unconsciously done every day. The restrictions in the recirculating duct and the limited area of the flues in the walls are accountable for many of the poor results obtained from furnace installations. That a furnace fails to perform satisfactorily is by no means an accident. When one knows what its limitations are and then sees what is sometimes attempted, it seems like an accident if a furnace works at all.

SOME FACTS ABOUT CIRCULATION STIMULATORS

When the occupant of a house finds it impossible to heat his place, he does a little experimenting on his own account, he then calls in the furnaceman who does some more, usually without producing any marked effect on the job. Finally somebody suggests that "the only way to make this job work is to put in a blower." *Usually that is the wrong way to go about it.*

Installing a blower for no other purpose than to overcome a defect by *force,* is an *extravagance.* Rightly applied, a blower will effect economies that cannot be duplicated in any other way, and they are not recommended for any other purpose.

Applying a blower to a furnace, should be for no other purpose than to remove the film of air in immediate contact with the heating surface more rapidly than it can be moved by gravity. As the film of air becomes heated, it carries away heat from the heating surface as it moves forward, and the more rapidly it moves, the faster the heat is removed because of a continuous film of cool air coming in contact with the hot surfaces. The capacity of a furnace which is operating under the most favorable conditions, can be increased from two to three times by using a blower. At Urbana this was demonstrated after alterations were made which had already increased the capacity by gravity circulation 1.8 times.

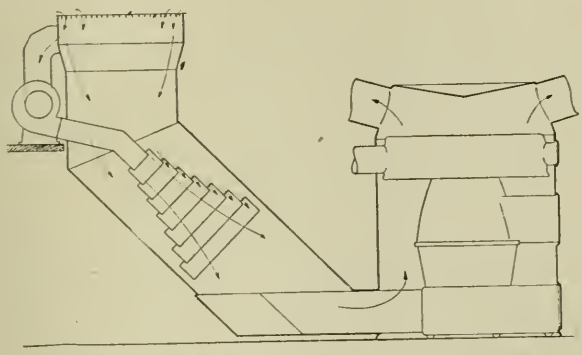

FIG. 5. DEVICE TO STIMULATE FLOW OF AIR FROM FURNACE

With a limited height of chimney and a reasonable limit to the flue gas temperature, the draft is fixed within rather narrow limits. The amount of draft fixes the rate of combustion, and the rate of combustion fixes the amount of heat obtainable from the fuel per unit of grate area. Therefore, with a fixed ratio of heating surface to grate surface, when the maximum heat transmission from a given amount of heating surface has been reached by gravity circulation, the only way to get more heat is to increase the air movement over the surface, by a blower device of some kind, or install a larger furnace with larger flues, etc.

The available pressure head is so slight in gravity work, that it needs but little resistance to stop the flow and sometimes completely reverse it. It does not require much additional pressure in many cases to set the air in motion and as soon as the ducts and flues become thoroughly heated, they will continue to operate satisfactorily.

Wherever long horizontal or crooked ducts prevail, a blower must be installed which is large enough to handle all the air that will be required to convey the maximum amount of heat into the building. Unless one is prepared to have such a blower operate continuously when heat is required, he may be disappointed in the results, as it becomes rather costly to pay for electric current as well as coal for continuous operation.

The reason why such a blower must operate continuously and not occasionally, as a circulator can be operated, is this: Some of the leaders offer very little resistance to the flow of air, whereas some of the others offer a great deal. Therefore, dampers must be inserted in such leaders offering but little resistance, so as to make them equal the resistance of the others. If this is not done, the bulk of the air will blow through the leaders having the least resistance, with the result that little would be gained in the delivery of more air or more heat to those rooms requiring a greater amount.

By making such adjustments of the dampers to equalize the flow, it introduces so much resistance to the whole plant that the furnace is unable to heat any part of it by gravity circulation alone. That is why the blower has to run all the time, when one is installed with the expectation that all of the air required by the building will have to pass through it.

All kinds of devices have been developed to stimulate the flow of air from furnaces without having to handle all the air required to properly heat a building. Many of such devices have long since passed out of existence, and have been forgotten. During the past two years, a new crop has sprung up, one of which, Fig. 5, was recently tested at the University of Illinois and reported to the last meeting of the *National Warm Air Heating & Ventilating Association.*

This particular device is intended to make it possible to heat a building by circulating air at low temperature instead of at high temperature, thus avoiding hard firing with its inevitable waste, repairs and renewals to the furnace. Higher efficiency can be obtained under all weather and operating conditions and less attention to the furnace will be demanded to get satisfactory results.

Tests made on this device show that at a register temperature of 120 deg. it is possible to induce 60 per cent more air to flow than by gravity and at the same time get an overall efficiency 20 per cent higher. As the register temperature rises, the gain becomes gradually less; thus, at 180 deg., the amount of air induced to flow over and above what would flow by gravity is 35 per cent more and the overall efficiency is but 3 per cent more. Please bear in mind that this gain was effected after alterations had been made in the furnace so as to get the best possible results by gravity circulation. The comparative results obtainable are shown in Fig. 4.

This circulator was designed to induce the flow of several times the volume of air that the blower of itself will handle. It, therefore, consumes so little current that the cost of the fuel and the electricity combined, even if the circulator is run continuously, amounts to less than the fuel alone would amount to when similar results are produced, if such a thing could be effected.

It offers no obstruction to the flow of air in the recirculating pipe, so that the furnace can be operated without the blower running just the same as though nothing was there, and the result will in no way be different. When more heat is wanted, the circulator can be started, and without opening the drafts, the same volume of heated air can be obtained as can be obtained by gravity at temperatures 60 to 80 deg. higher. If

the same velocity could be produced by gravity as with the circulator at low temperatures, this device would be unnecessary. The only other way to accomplish corresponding results, would be to make the leaders large enough to convey the required volume at the low temperatures. In most cases this would be impracticable for lack of space, and besides, the efficiency of a furnace falls off so rapidly under such conditions that nothing like the same economy could be attained as with the circulator.

Some day circulators of some kind will be considered as necessary as any other part of a furnace plant. They will not be looked upon as something to fall back upon to make a defective plant acceptable, but they will be considered as an economical necessity, by means of which air can be circulated at more healthful temperatures, regardless of weather conditions.

The work done by Professor Willard and his staff at the University of Illinois, under the direction of the *National Warm Air Heating & Ventilating Association,* is a wonderful accomplishment. It means more to the future success of the furnace business than most of those who have had direction of this work fully realize. In the adoption of the code, herein referred to, a foundation has been laid for progress and improvement. As previously stated, the code is not perfect, but with a good foundation on which to build in the future, amendments can be made to correct and improve the code as progress and further knowledge seems to indicate.

Better furnaces will undoubtedly be designed, as they are certainly needed. None of the furnaces on the market are what would be considered a first-class design for a convector. Too much of the heat generated in a furnace is now wasted by radiation. There is no reason why a furnace cannot be designed which will show an efficiency ranging between 80 and 90 per cent. There is no furnace on the market today which will show better than about 60 per cent.

It is estimated that there are about 10,000,000 homes without furnaces in this country in climates where heat is necessary, and it is also estimated that 80 per cent, or 8,000,000, of these homes could be equipped with furnaces so that if this business could be developed it would amount to over two billion dollars, not including replacements. Thus one gets an idea of the importance of the work being done to standardize furnace practice. It has been further estimated that if those who have inefficient and ancient equipment could be induced to install modern furnaces it would be sufficient to keep the present industry busy for five years.

Owing to the poor success so frequently attending the installation of warm-air furnaces in homes of moderate size the furnace business has gradually gotten down to the stiffest kind of competition and about the only thing that recommends it to the average man is its low cost in comparison with steam or hot-water heat. Undoubtedly if it is put on a more scientific basis better installations will be the result and the public will gather confidence and will be willing to pay more money for a properly installed furnace, as it has many attractive features to recommend it. In some classes of buildings it is preferred to the complicated steam or hot-water systems and the expense necessary to install this type of apparatus.

DISCUSSION

V. S. DAY: I have one or two comments to make on Mr. Still's paper. He mentions that the University of Illinois had no published data on the sizes of wall stacks. He mentioned also that I had called his attention to the fact that this was slightly in error.

On page 133 of Bulletin 120, the same one to which Mr. Still has referred this morning, a table is published giving the relative carrying capacities of two or three sizes of wall stacks when attached to the same leader pipe.

The results of those tests are about as follows: If the wall stack is three-fourths the area of the leader pipe, it will handle about nine-tenths of the heat carrying capacity. If it is reduced to 50 per cent of the leader pipe size, it will handle only about 75 per cent of the heat carrying capacity. If any of you desire to see a copy of that bulletin I have one here. It may be of some value to you.

Mr. Still also discussed the significance of the work that we have done at Urbana and it has shown, as the material indicates, that the factor 9 in the formula for leader pipe sizes is not quite big enough, unless you are willing to run into high register temperatures. Our recommendation was based on register temperature of 175 deg. You can't go much below 175, for the reason that if you desire giving heat to your room you will have to handle more air if you do not increase it to a higher temperature and that means higher velocity and consequently higher resistance in the pipes. So 165 to 170 deg. you may say is the minimum design temperature which might prevail at the register; 195 seems to us extraordinarily high.

If any one wants to see a copy of the report mentioned as not yet being in print, I have a mimeograph copy here, and those of you who are particularly interested, may secure copies by addressing the University of Illinois or the secretary of the *National Warm Air Heating and Ventilating Association.* A few copies are available, also Bulletins 112, 117 and 120, bearing on the subject of warm air furnaces.

J. D. HOFFMAN: Permit me to commend the paper presented by Mr. Still in that he has shown accurately the field of the warm air furnace. It is a large field. I want to refer to one point in Mr. Still's paper that I did not quite understand. In the last sentence of paragraph 1, page 319: "It was further demonstrated that the furnace is affected much more by any restriction to the flow of air in the recirculating duct, than by a similar restriction in the leaders from the furnace to the registers."

V. S. DAY: I think I can explain Mr. Still's statement in the paper. At Cleveland we did give a report bearing on the effect of register grilles in discharging air from a warm air furnace system. And the statement was made there that you could take the register grilles out and it had no noticeable effect. So it appears that no register in the hot side had any effect on the capacity; it did have a slight effect on the cold side.

No. 639

THE DERIVATION OF TRUE THERMAL CONDUCTIVITY COEFFICIENT FROM OVERALL TEST RESULTS

Report of Cooperative Work by A. S. H. & V. E. Laboratory and the U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.

P. Nicholls, Pittsburgh, Pa.
Member

The thermal conductivity coefficient of most materials varies with and has a definite value for each temperature. The values for metals are generally so expressed, but those for poor conductors are less accurately stated. When the method of test only gives an overall value of heat flow for an appreciable thickness of the material, the coefficient deduced is only an average for the temperatures at the surfaces.

The paper develops a method for deriving a curve of conductivity coefficient against temperature, and shows how such values can be applied to facilitate more accurate and simpler computations of applications of the material to practical conditions.

THE term, thermal conductivity coefficient, which is usually expressed shortly as Heat Conductivity, has been rather loosely used in engineering literature and reports of investigations dealing with poor conductors and insulators. It will here be employed in its correct meaning of the number of heat units transferred by conduction, per unit area, across unit thickness, per unit time, per degree difference of temperature between the faces, the direction of heat flow being perpendicular to these faces.

The majority of tests for heat transmission is based on establishing constant flow and temperature conditions. The test sample has usually been a spherical shell, a cylindrical shell, or a flat block with parallel faces. In the two latter cases the cylinder or plate is assumed infinite, or test conditions imposed so as to make the lines of flow of heat perpendicular to the axis or faces respectively.

The result is then expressed in the form $H = S.C\ (T_1 - T_2)$
where $H =$ total flow per unit time.
$C =$ thermal conductivity coefficient.

Paper presented at the Semi-Annual Meeting of American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

S = shape factor which fixed by the geometrical form and dimension of the test sample.
T_1 = inner surface temperature of the face.
T_2 = outer surface temperature of the face.

The method of test being such that these latter temperatures are uniform over each surface or at least would be if perfect symmetry of the materials and conditions existed.

The value of C obtained by such a formula assumes that the conductivity is constant through the material, and since this is not the case, particularly for materials with a large percentage of air spaces, it gives what may be called an average of the conductivity for the temperature limits, T_1 to T_2.

If values are found for H, for different values of T_1 and T_2, the resulting curve is that of average conductivity against temperature difference, usually showing an increase of the coefficient with an increase of temperature difference.

Although such values are sufficiently correct for field requirements, especially when the investigations have not been continued so as to connect the values with variation in the material, yet it is well to try and carry the analysis so as to express the coefficient as a function of the cause of the change, and a method of deriving true values is given in the following.

Within certain limitations that will be given, there is no reason to suppose that the conductivity coefficient at a given point of the material will be affected by anything else than the temperature at that point, and it can, therefore, be written as:

$$C = \text{a function of the temperature} = f(t) \qquad (1)$$

It is first necessary to set certain limitations on the material and conditions of test so as to establish a basis for the deductions made. These necessary limitations are as follows:

a The material must be homogeneous relative to the lines of heat flow of the particular test. It need not have the same type of homogeneity in all dimensions, but, for instance, if it is laminated, then all the lines of heat flow at any portion of their length must make the same angle with each lamination.

b Heat must be transferred only by the material or its entrained air; that is, not by air filtration through the material. This limitation is not necessary, however, if the same set of conditions recur at each temperature in any element of the material wherever it is located, and to make it perfectly general and the values applicable to other shapes—if the filtration per unit area is also independent of the thickness or shape.

c That each element of the material shall have a certain definite and identical state corresponding to each temperature, and shall recur to that state at each recurrence of that temperature. For example, if the material is hydroscopic, then each element shall hold a definite amount of moisture corresponding to a definite temperature, independent of where the element is located. It could be imagined that certain change of shape or size of an element would be produced by a definite temperature, and that this might affect the conductivity. Then the same change should occur in any element and also any changes should be independent of the shape of size of the whole block of material.

d As an alternative to (*c*), *even if the material is permanently changed by any or all increases in temperature,* a true value of $C = f(t)$ may be obtained from a series of tests in which the temperature is raised by successive steps from

the lowest to the highest temperature of the hot side, and if no test results are used in which the hot temperature was lower than that of the previous test. To visualize this limitation, it can be imagined that the material is permanently damaged to a certain extent, and that the maximum temperature to which it had been heated is a measure of the damage, and that such damage causes a change of conductivity in addition to the natural change due to temperature. It would follow that if this material were now tested at a lower temperature, its conductivity would not be the same as it was before heating to the higher. As a corollary to this it follows that the $C = f(t)$ thus found for such a material can only correctly be applied to predict the heat flow when the hot temperature is not less than the highest it has been subjected to. For example, it could be used for a steam pipe condition, but not rigidly correctly for a furnace wall at temperatures lower than that of the hottest fire which ever has occurred.

Assume tests have been made by any method on any shape of material, for example a spherical or cylindical shell or a flat block by any method involving establishing constant flow of heat at constant temperature conditions. Let Fig. 1 represent a section through the test piece, which for illustration is taken as a sphere or cylinder. The higher temperature and the source of heat is assumed to be in the interior.

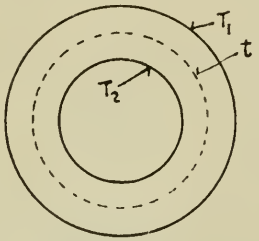

FIG. 1. SECTION THROUGH A SPHERICAL OR CYLINDRICAL TEST PIECE WITH INTERIOR SOURCE OF HEAT

Consider for the time being only the observations of:

H = total flow of heat per unit time under constant conditions,
T_1 = the temperature of the inner surface,
T_2 = the temperature of the outersurface,
and let t = temperature of any intermediate layer (not observed),
S = the shape factor.

Although illustrated for a cylinder, these values can be assumed to be for any other shape, the only limitation being that the layers of T_1, T_2 and t are equitemperature surfaces.

Let a series of constant conditions be established and readings taken; a series of values for H, T_1, and T_2 will be obtained. If this were done by running up and down, that is by first increasing T_1 by successive steps and then decreasing it from the highest temperature, and if the values were found to be the same, it would be some proof of the constancy of the material.

From these values let a curve M be plotted showing H against T_1 and another, N, showing T_1 against T_2. Such curves are shown in Fig. 2.

Curve M can be expressed as $H = F_1(T_1)$, (2)

Curve N can be expressed as $T_2 = F_2(T_1)$, (3)

where F_1 and F_2 are some unknown functions of T_1.

There is also the relation of (1) of $C = f(t)$, and by considering an elemental ray of heat and integrating over the whole body:

$$H = S \int_{T_2}^{T_1} f(t)\, dt. \tag{4}$$

or,

$$F_1(T_1) = S \int_{F_2(T_1)}^{T_1} f(t)\, dt. \tag{5}$$

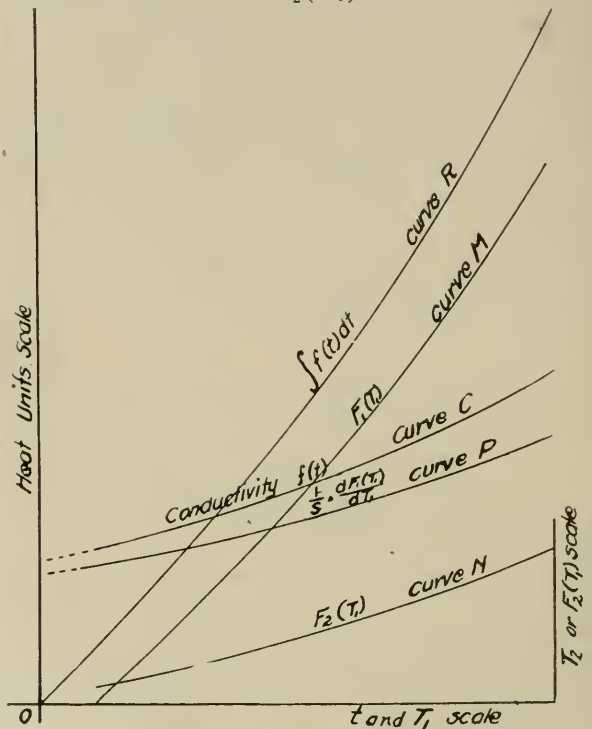

FIG. 2. CURVES OF CONDUCTIVITY-TEMPERATURE RELATIONS

The values of S for the particular shapes are:

$$\text{For a flat block, } S = \frac{A}{B}. \tag{6}$$

where A = area, and B the thickness.

$$\text{For a cylindrical shell, } S = \frac{2\pi L}{\log_e \frac{R_2}{R_1}} \tag{7}$$

$$\text{For a spherical shell, } S = \frac{4\pi R_1 R_2}{R_2 - R_1} \tag{8}$$

where R_1 and R_2 are the inner and outer radii in both cases, and L the length of the cylinder.

All these values and those of the other equations are based on one and the same unit value being used for each length and time, and care is necessary when using the mixed units common in engineering practice.

Equation (5) shows a general relationship expressed in terms of $f(t)$ and T_1; it is desired to find $f(t)$ from this. As the algebraic expressions for curves M and N, Fig. 2, are not known it is necessary to find a graphical solution.

Equation (5) can be expressed as:

$$\frac{1}{S} F_1(T_1) = \int^{T_1} f(t)\,dt - \int^{F_2(T_1)} f(t)\,dt.$$

Differentiating both sides with respect to T_1,

$$\frac{1}{S}\frac{d\,F_1(T_1)}{dT_1} = \frac{d}{dT_1}\int^{T_1} f(t)\,dt - \frac{d}{dT_1}\int^{F_2(T_1)} f(t)\,dt$$

$$= f(T_1) - \frac{dF_2(T_1)}{dT_1} \cdot \frac{d}{dF_2(T_1)}\int^{F_2(T_1)} f(t)\,dt$$

$$= f(T_1) - \frac{dF_2(T_1)}{dT_1} \cdot f\left[F_2(T_1)\right]$$

$$\therefore f(T_1) = \frac{1}{S} \cdot \frac{dF_1(T_1)}{dT_1} + \frac{dF_2(T_1)}{dT_1} \cdot f\left[F_2(T_1)\right] \qquad (9)$$

By finding a value for the right hand side of (9), the value of $f(t)$ at T_1 will be obtained.

As $\dfrac{dF_1(T_1)}{dT_1}$ is the tangent to curve $F_1(T_1)$, curve M in Fig. 2, at the point T_1, by graphically finding the tangent at successive points on M and plotting the values of $\dfrac{1}{S} \cdot \dfrac{dF_1(T_1)}{dT_1}$ to a desirable scale, a new curve P is obtained.

Considering the second term of the right hand side of equation (9), $\dfrac{dF_2(T_1)}{dT_1}$ is the tangent to the curve N where it cuts the ordinate at T_1, and this value is to be multiplied by $f[F_2(T_1)]$. This latter is the value of $f(t)$ when $t = F_2(T_1)$; that is T_2, and as the curve $f(t)$ has not yet been drawn it is not known.

In Fig. 3, curve $\dfrac{1}{S} \cdot \dfrac{dF(T_1)}{dT_1}$, or curve P of Fig. 2, is shown. Assume that the curve C is the desired $f(t)$ curve. On the T_1 axis

take the point T_1. Let T_2 be the cold surface temperature corresponding to T_1 for the hot. Let T_3 be the cold corresponding to the hot T_2, and so on. In any practical method of test there is a limiting temperature T_0, that of the air surrounding the colder surface, say, at which hot temperature = cold temperature = T_0, and if the hot temperature were carried lower the flow of heat through the material would be in the opposite direction. (Note that the value of the cold temperature T_2 for any hot temperature can be read off from the curve N, Fig. 2.) In Fig. 3 draw the ordinates through T_1, T_2, etc., then A_2A_3 is equal to the expression $\frac{dF_2(T_1)}{dT_1} f[F_2(T_1)]$, $B_2B_3 = \frac{dF_2(T_2)}{dT_2} f[F_2(T_2)]$, $C_2C_3 =$ etc.

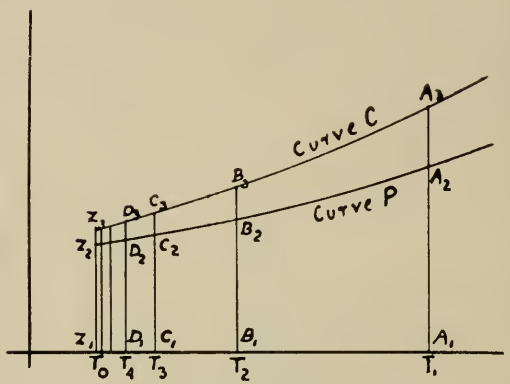

FIG. 3. CURVE C SHOWING TRUE CONDUCTIVITY COEFFICIENT AGAINST TEMPERATURE

Using k_1, k_2, etc., to represent $\frac{dF_2(T_1)}{dT_1}$, $\frac{dF_2(T_1)}{dT_2}$, etc., respectively,

then $A_2A_3 = k_1B_1B_3 = k_1(B_1B_2 + B_2B_3)$,

and $B_2B_3 = k_2C_2C_3 = k_2(C_1C_2 + C_2C_3)$

till $Z_2Z_3 = k_z(Z_1Z_2 + Z_2Z_3)$

or $Z_2Z_3 = \frac{k_z}{1 - k_z} \cdot Z_1Z_2$; therefore

$$A_2A_3 = k_1\left\{B_1B_2 + k_2(C_1C_2 + k_3(D_1D_2 + \text{---} + k_2(\frac{Z_1Z_2}{1 - k_2})\right\} \quad (10)$$

In practice it is only necessary to run (10) to a few terms as the products of the k's rapidly reach a negligible quantity.

By this means the actual values of the second term of (9) can be found for various values of T_1 and added to curve P of Fig. 2, give curve C, which shows the true conductivity coefficient $f(t)$ against temperature.

In practice on the curves can safely be extended a little at the lower end so as to make the result more complete, and this can in any case be done to T_0, the air temperature, say, but extension at the upper end would be more doubtful.

From curve C one for $\int f(t)\mathrm{d}t$ can be obtained by any graphical integrating method and is shown in Fig. 2 by R. Curves C and R give complete data of the conductivity for the assumed material within the temperature limits tested, and these should be independent of the method of test. A solution of any heat flow problem for this material when the outer temperature of the surfaces are known can be obtained from equation (4), $H = S\int_{T_2}^{T_1} f(t)\mathrm{d}t$, by reading the difference between the lengths of ordinates of curve R, at T_1 and T_2, and multiplying by S.

The curve of temperature through the material can also be plotted for any constant condition of heat flow H, with hot temperature T_1, say. This is done by assuming some lower temperature t, obtaining from curve R the value for $\int_t^{T_1} f(t)\mathrm{d}t$, and substituting this and H in the equation $H = S'\int_t^{T_1} f(t)\mathrm{d}t$. This gives a value for S', from which the location of temperature t can be found. Suppose it is a cylindrical shell, then from (7) $S' = \dfrac{2\pi L}{\log_e \dfrac{R_1}{r}}$ and R_1 is known, being the location of T_1. Then r will be found as the location of the particular t chosen. Thus by taking successive values of t between T_1 and T_2, the curve of temperature across the section can be plotted.

If the tests are made under uniform conditions so that smooth curves result, and if the results are concordant so as to give a definite curve for $F(T_1), -M$ in Fig. 2 —, the graphical work can be done very accurately and a final check made by the agreement of the values of H as calculated from the integral curve R with the test results as shown by curve M. If curve M does not pass through all the test points, but is merely a mean value, with some of the points lying considerably off it, the result is only as accurate as the guess in drawing the M curve.

The method will, however, indicate the relationship of the conductivity coefficient to the temperature more intelligently than does the plotting of it against temperature difference, whether this be that between the hot surface and the air as is sometimes done, or between the outer surfaces of the test piece. Both these methods will give a coefficient which is dependent on the thickness of material tested, since, expressed in the notation here used, it will be $\dfrac{1}{T_1 - T_2}\int_{T_2}^{T_1} f(t)\mathrm{d}t$ plotted against

$(T_1 - T_0)$, where T_0 is the air temperature during the test, or against $(T_1 - T_2)$, either of which methods will show a conductivity coefficient which will be different for each thickness tested, as can be seen by taking such results from the curve in Fig. 2.

If the method of test is such that one of the faces of the test piece is kept at the same temperature while that of the other is varied, equation (3) becomes $F_2(T_1) =$ constant, and curve N in Fig. 2 will be a line parallel to the T_1 axis, and $\frac{dF_2(T_1)}{dT_1} = 0$. As $f[F_2(T_1)]$ remains a real value, equation (9) becomes

$$f(T_1) = -\frac{1}{S}\frac{dF(T_1)}{dT_1} \qquad (11)$$

so that the $f(t)$ curve will now be the P curve in Fig. 2.

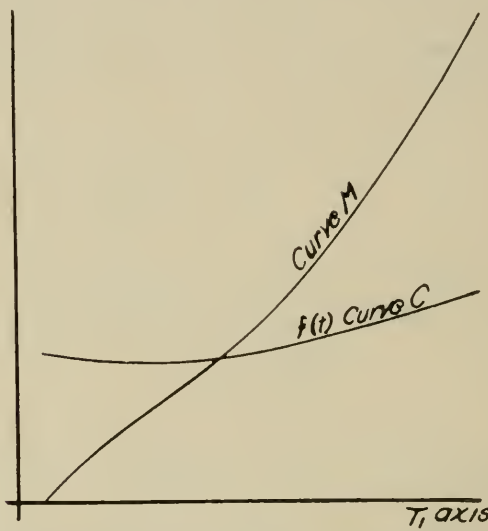

FIG. 4. POSSIBLE CONDUCTIVITY CURVES ILLUSTRATING RISING CONDUCTIVITY AT LOW TEMPERATURE END DUE TO HYGROSCROPIC MOISTURE

It would be expected that the conductivity-temperature curves for materials containing considerable entrained air would usually be of the general form as shown by C in Fig. 2, this rising form being due to the increasing conductivity of air with temperature, and also to the increase in the transfer of heat due to thermal radiation across the small air spaces.

It is very probable that some of these same materials will have conductivities of the form shown in Fig. 4, the rising conductivity at the low temperature end being due to hygroscopic moisture. Such a result will be brought out if the curve M is truly drawn to the test results.

The curves for test results as shown in Fig. 2 are imaginary, to illustrate the principles. An actual illustration is given in Fig. 5. This is from a test made at the Mellon Institute on 85 per cent magnesia.

Acknowledgment is due to Dr. E. R. Weidlein, Director of the Institute, for permission to use their data, and the work from which it is taken was done by the Fellows M. G. Mason and R. H. Heilman.

The apparatus used has been described in papers by the Fellows, and in this case consisted of a standard 3-in. iron pipe electrically heated and on the guard ring principle. This particular sample was 3.01 in.

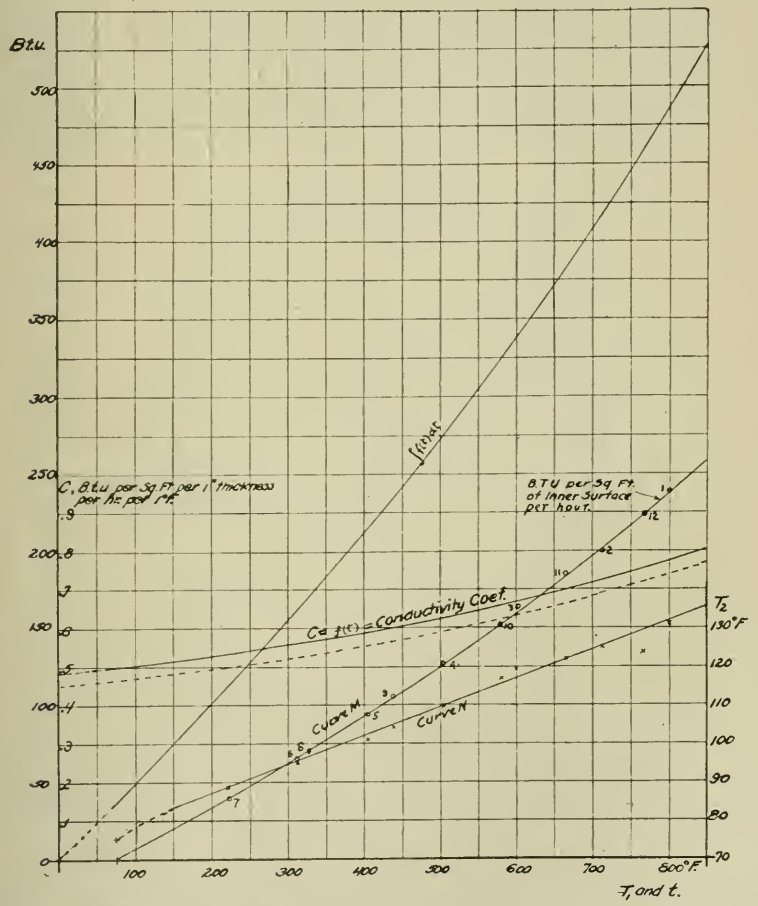

FIG. 5. CONDUCTIVITY-TEMPERATURE CURVES OBTAINED IN TESTS OF 85 PER CENT MAGNESIA

thick as a single layer, and split horizontally, the inner radius being 1.8 and the outer 4.81 in. It was suspended horizontally, and the air temperature during the test varied from 74 to 78 deg. fahr.; such a test takes several weeks due to the time required to reach constant conditions for each change in pipe temperature. During 50 per cent of the time it varied from 74 to 76 deg. fahr. To obtain uniform conditions the results in each case have been reduced to a basis of 75 deg. fahr., thus fixing a zero point for the two experimental curves.

The sample before test weighed 23.75 lb. and being 36 in. long, thus had a specific weight of 18.15 lb. per cu. ft.

The test points fit very closely to the mean curves with the exception of point 12, which, being the last, possibly was not allowed sufficient time to reach constancy. The $F_2(t)$ curve is a straight line except at the lower end when it has to be curved down to reach the zero point of $T_1 = T_2 = 75$ deg. fahr.

The conductivity and integral curves were derived in the method previously outlined. Such a material, as far as one sample is concerned, is homogeneous due to its method of manufacture and the resultant conductivity-temperature curve so derived should fairly well meet the limiting conditions previously given. It is to be noted, however, that the

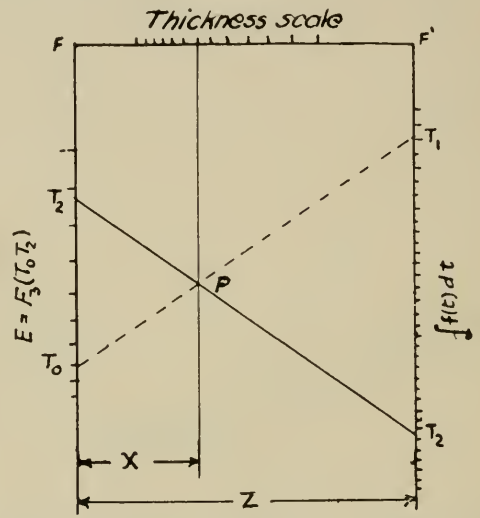

FIG. 6. ALIGNMENT CHART TYPE METHOD FOR HEAT TRANSMISSION OF INSULATED HOT BODY IN AIR

tests were started at the highest temperature, and if any permanent changes occur which affect its conductivity, then such are masked by the method of test.

Having the integral curve for the conductivity, it is possible to devise a graphical method for solving transmission problems. Such problems, for exact solution, are tedious when treated algebraically, as the surface transmission is again a function of the temperatures, and can be expressed as:

$$H = A \times F_3(T_2 T_0) \tag{12}$$

where A = area of the surface,

$F_3(T_2 T_0)$ = surface transmission of unit area with surface temperature T_2, and air temperature T_0.

The transmission from an insulated hot body, with the outer surface of the insulation exposed to the air, and with known temperatures T_1 and T_0 of the body and air respectively, would be given by:

$$H = A \times F_3(T_2T_0) = S\int_{T_2}^{T_1} f(t)\,dt$$

Probably the most convenient method is that which is shown in principle in Fig. 6, which is of the alignment chart type. The $\int f(t)\,dt$ is plotted on one of the parallel vertical lines, and the value of $F_3(T_2T_0)$ on the other. For general purposes it is convenient to plot them to the same scale, both being heat units per unit area per unit time per degree temperature.

Take a line such that $\frac{X}{Z-X} = \frac{S}{A}$;

$$\text{that is, } X = \frac{SZ}{S+A} \tag{13}$$

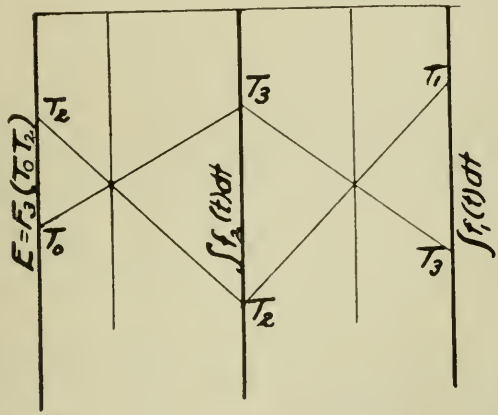

FIG. 7. THE ALIGNMENT CHART TYPE METHOD APPLIED TO COMPOSITE WALLS

Join the air temperature T_0 to the hot temperature T_1. On the point P swing a straight line to find its position where its intersections with the two scale lines give the same temperature T_2. Then T_2 is the temperature of the outer surface and the heat flow is

$$H = (\text{heat units represented by length } T_0T_2) \times A$$
$$= (\text{heat units represented by length } T_2T_1) \times S$$

Plotting a scale for $X = \frac{SZ}{S+A}$ along FF' completes the diagram.

The use of such a diagram can be extended to include changes of $\int f(t)\,dt$ and $F_3(T_2T_0)$ with other variables, such as specific weight of the material and area of the surface respectively, by plotting other scales parallel to them.

The only shapes of practical interest are flat blocks and cylindrical shells. For these equation (13) becomes,

For flat parallel blocks, $X = \dfrac{Z}{B+1}$

For cylindrical shells $X = \dfrac{Z}{1 + R_2 \log_e \dfrac{R_2}{R_1}}$

The FF' scale can therefore be plotted directly against B, and the other value can be connected by additional alignment charts.

The same method can be extended to composite walls, as shown in Fig. 7, but in this case a solution is by trial and error to obtain from T_1 and T_0 the points T_2 and T_3 which will give intersections on the two intermediate axes.

Fig. 8 shows a chart for transmission through 85 per cent magnesia, based on the conductivity value shown in Fig. 5. To avoid too small a scale reduction in the printing the auxiliary chart for the shape factor (S) for pipes has been omitted and a scale for $D_2 \log_e \dfrac{R_2}{R_1}$ given on the bottom line. The surface transmission values are from tests of the Mellon Institute with 6-inch outside diameter. The surface transmission from cylinders actually varies with the diameter and Mr. Heilman (*Mechanical Engineering*, May, 1922) gives a formula of $(T_2 - T_0) = \dfrac{272.5\,h}{h + \dfrac{564}{D^{0.19}}}$, h being the B.t.u. per hour per square foot and D the outside diameter of the cylinder in inches, this being for horizontal positions. Additional side scales could be plotted to cover a relationship like this, but for clearness are omitted.

CONCLUSION

The accuracy of constants is no greater than the tests from which they are derived. The truth of the values obtained by applying such constants in the computation of actions in practice is dependent on the material used being the same as that tested, or in the ability to define and obtain the desired quality. As a consequence such analyses as are suggested in this paper do not ensure greater accuracy unless the same refinement is used for other influencing conditions.

However, the investigator is responsible for eliminating all possible errors or approximations in interpreting the results of his tests, and in any case the recognition of and interpretation on the true fundamental basis in itself keeps the viewpoint clearer and is more likely to produce results which can be coordinated with those of others, and to avoid the necessity of a repetition of the tests in the future.

This paper has not included a discussion of the more direct method of obtaining the true conductivity by taking temperature readings through

the material, along the lines of heat flow. This is possible and has in effect been done, though the exact temperatures are hard to obtain in most cases, due to the higher conductivity of the thermocouple wires and the upsetting influence of the necessary holes.

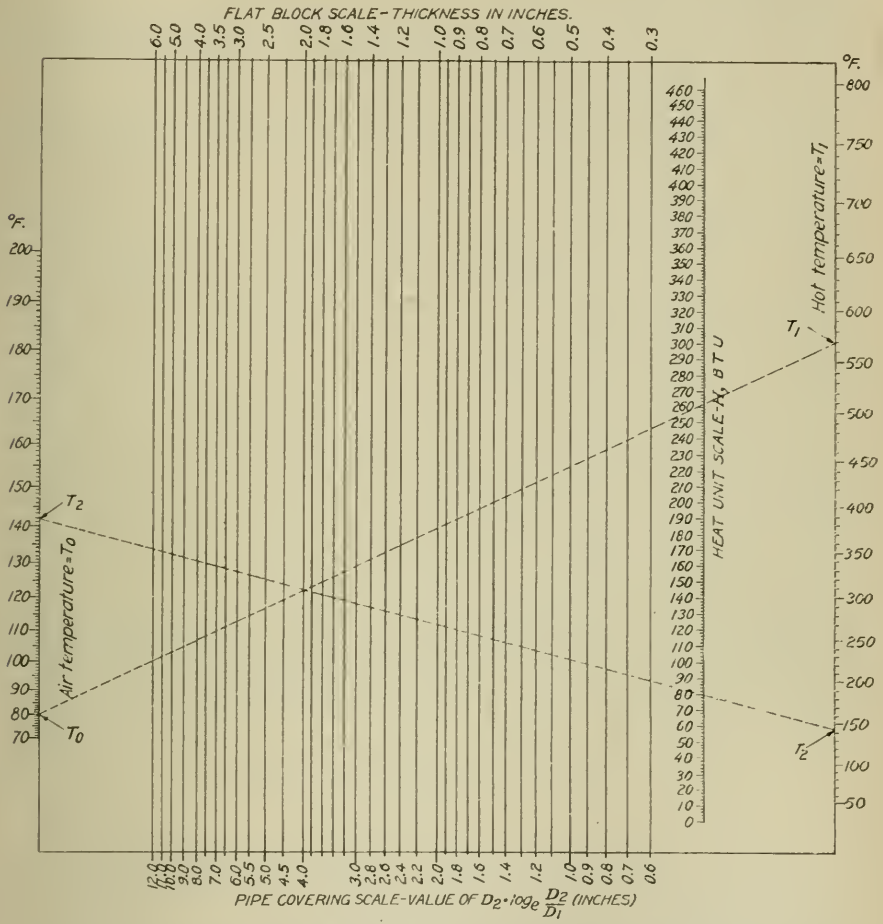

FIG. 8. A CHART FOR TRANSMISSION THROUGH 85 PER CENT MAGNESIA BASED ON CONDUCTIVITY VALUES SHOWN IN FIG. 5

USE OF DIAGRAM

Flat Blocks—Joint air temperature T_0 (80 deg. fahr. say) to hot surface temperature T_1 (570 deg. fahr. say). On its intersection with the thickness of insulation (2 in. say) swing a straight edge until T_2 reads the same on both side scales; T_2 (142 deg. fahr.) is temperature of outer surface of insulation and length T_0T_2 transferred to scale H (124) is B.t.u. per sq. ft. per hour lost through insulation.

Pipe Covering—If D_1 is outside diameter of pipe, and D_2 outside diameter of covering, both in inches, find value of $D_2 \log_e \frac{D_2}{D_1}$, proceed as for Flat Blocks and T_2 will be outside temperature of covering and heat loss per foot run per hour is the length T_0T_2 transferred to scale H multiplied by $0.262D_2$.

DISCUSSION

L. L. BARRETT (written): This paper is a timely presentation of a subject which, unfortunately, seems to be very imperfectly understood by many engineers. The paper is most useful in clarifying the general conception of conductivity and it is to be hoped that it will have a beneficial effect in this direction. Only recently have conductivities been plotted as functions of temperature difference. The procedure rests on no rational basis and is bound to give incorrect results.

It seems desirable to point out another and much simpler method whereby conductivities can be obtained from test results. Future investigators can then see the two methods side by side and take their choice as to the method to adopt.

It can be easily shown that if the conductivity curve as a function of temperature is a straight line between the temperatures corresponding to the two surface temperatures measured in any particular test, the heat loss obtained divided by the product of the shape factor and temperature difference gives a point on the true conductivity curve of the material at a temperature equal to the mean of the two surface temperatures.

This proposition in a less general form was advanced by Dr. Carl Hering some years ago but does not appear to have been utilized in subsequent well known investigations. As the proof of the proposition is little known, but very simple, it seems well to give it so that subsequent investigators who refer to the discussion of this paper will be able to make an intelligent choice between the two methods. We will use the same nomenclature as in Mr. Nicholls' paper.

Assume that conductivity is a linear function of temperature between the temperatures corresponding to the two surface temperatures obtained in any single observation of T_1 and T_2. By suitable experimental methods these temperatures can be made quite close together.

Then, $C = a + bt$ (1)

$$\text{and,} \quad H = S \int_{T_2}^{T_1} (a + bt)\,dt \tag{2}$$

$$H = Sa(T_1 - T_2) + \frac{bS}{2}(T_1^2 - T_2^2) \tag{3}$$

Let K be the value of the conductivity obtained from any set of observed values of H, T_1 and T_2 by means of the formula

$$K = \frac{H}{S(T_1 - T_2)} \tag{4}$$

$$H = SK(T_1 - T_2) \tag{5}$$

Equating values of H from (3) and (5) and dividing through by $T_1 - T_2$ we obtain

$$K = a + \frac{b}{2}(T_1 + T_2) \tag{6}$$

Comparison of (1) and (6) shows that K, the conductivity found directly from test results by means of the usual formula $K = \frac{H}{S(T_1 - T_2)}$, is the true conductivity at temperature $\frac{1}{2}(T_1 + T_2)$. It will be noted that the shape factor cancels out in the solution, thus proving that the conductivity obtained is independent of shape.

Points on the true conductivity curve of any material can thus be obtained directly from experiment. Even though the conductivity curve considered as a whole is a curve and not a straight line, the points obtained, and therefore the curve, will be correct providing the temperature difference used in the determination of each point is kept low. This is readily done even at the higher temperatures by placing an outer insulating covering over the covering being tested, thus cutting down the temperature drop in the covering under test.

Even when the temperature difference between the two sides of the covering is comparatively great as in the experiments from which the first two curves on Fig. 5 were drawn, the conductivity may be plotted against the mean of the two surface temperatures and give a conductivity curve of considerable accuracy. It is significant that when the data given by these experimental curves is replotted in accordance with the method set out in my remarks the resulting curve can be distinguished only with difficulty from the author's conductivity curve based on the same data. The mean variation between the two curves is less than 0.01 B.t.u. There can be no question but that the advantage of simplicity is on the side of this method of obtaining conductivities direct from test results. It does not require the drawing of tangents or the use of convergent series or other mathematical approximations. The method is believed to be fully as accurate as the author's method because it eliminates these complications, and because it is mathematically exact. The only approximation is the physical approximation in the linear conductivity assumption. For most of the materials we are considering and with comparatively small temperature differences, the error involved disappears.

The complexity and possible source of error in the author's method lies in the fact that the increment which must be added to the P curve to obtain the conductivity curve is equal to the sum of a number of terms, each of which is the product of two or more tangents of angles, and the further fact that the P curve itself is a curve of tangents. Due to the rapid variation of the tangent and the difficulty of determining the exact point of tangency, it is believed that these errors will count up more rapidly than will any error which is made in assuming a linear conductivity relation over a small temperature range.

The converse proposition should also be stated. Having the conductivity curve as a function of temperature for a particular material, the conductivity to be used in working out any problem is simply taken off the curve at the mean of the two surface temperatures. This is true whatever the thickness of the insulation.

Finally, it should be understood that in order to obtain good conduc-

tivity curves by either method it is necessary that smooth curves be drawn through the experimental points to obtain the M and N curves.

L. B. McMILLAN (written): The author has presented an unique method of calculating conductivities at any temperature from tests results and it is of especial interest to note that the method gives the conductivity at any temperature regardless of the fact that the given temperature is only one of the many temperatures which exist in the insulation in any set of test results. Therefore, this is to be considered as a most important contribution in connection with heat insulation calculations.

The principal difficulty seems to be in the solving of equation 9, which requires graphical methods. The necessity of evaluating tangents to two experimental curves introduces a considerable source of error because of the difficulty in accurately drawing such tangents.

The example illustrated in Fig. 5 would have been much more easily followed had numerical examples been given, showing how the values of the P and C curves were obtained. For example, it is not stated whether the B.t.u. scale represents B.t.u. per sq. ft. of pipe surface or B.t.u. per linear foot of pipe and the time unit is not given. Apparently, this scale represents B.t.u. per sq. ft. of pipe surface, per hour, but if this is true then a different value of S than that given in equation 7 must be used. The equation for S in this case should be $S = \frac{1}{r_1 \log \frac{r_2}{r_1}}$

In calculating a point for the C curve in Fig. 5, even if equation 10 is carried to only two terms, four tangents must be measured, three of which are below the limit of the experimental data and if carried to more terms the proportion of tangents required at low points is still greater.

It is apparent that the N curve in Fig. 5 should not be a straight line. The necessity of the curved dotted portion to reach the 75 deg. point illustrates this. In most cases, the N curve will be concaved downward and may be very considerably curved in many cases. Taking it as a straight line simplifies the measurement of the required tangents and in this case probably results in but little error. However, it should not be assumed that this is a straight line relationship or in other words it should not be assumed that the temperature of the outer surface of insulation varies directly with the inside surface temperature.

Furthermore, the effect of downward curvature of the N curve would tend to offset the opposite curvature of the P curve and, therefore, would tend to straighten out the curve of conductivities, making it more nearly a straight line.

This is of especial importance in view of Mr. Barrett's proof that in case the conductivity curve is a straight line the true conductivity at the mean of the two surface temperatures is the same as the overall conductivity calculated directly from actual test results. This gives a more direct method of measuring true conductivities and one which should be very accurate. This is true even for materials whose conductivity curves

are not exactly straight lines; because, for such materials, the error in assuming that the conductivity curve is approximated by a straight line over a small temperature range is entirely negligible.

In this connection, it may be of interest that the writer has made many tests, the results of which indicate that, for materials of high insulating value, the conductivity curves are either straight lines or vary only slightly from straight lines over relatively high ranges of temperatures.

It is noted that, referred to Fig. 8, the author states that the surface transmission values used were taken from tests on a surface of 6 in. outside diameter. Presumably the same surface transmission factors were used for both flat and cylindrical surfaces.

In this connection, if Mr. Heilman's equation, quoted by the author, is applied to flat surfaces where D = infinity, $(T_2 - T_0)$ will be shown to equal 272.5 regardless of the value of h. This is obviously impossible and therefore this equation is not one which can be applied generally to conditions beyond the range of the tests on which it is based.

The Author (written): I am aware of and have used Doctor Hering's method as given in his paper Vol. XXI *Trans. Elec. Chem. Soc.*, 1912, p. 511, and the paper and the discussion were a valuable contribution. However, as Mr. Barrett states, it is based on assuming that the conductivity coefficient-temperature curve is a straight line, and we should hardly be content to assume that this is true except for small values of $T_1 - T_2$. To obtain such small values by test would entail a great increase in cost in the making of them, and the introduction of a thick layer of material over the test sample, or the use of cooling methods to maintain any desired temperature for T_2 would mean added expense and the possibility of bringing in disturbing factors.

Using his method on the test results from which Fig. 5 was developed, would give a C curve up to $T_1 = 465$ deg. fahr. only, and the rest of the curve would have to be drawn by guess. This could be done and the integral curve derived, and its correctness tested by using it to calculate points on the test curves, and then by trial and error changing its shape until agreement is obtained at all points. Such a procedure is long and not very satisfactory.

One of the disappointing features of many investigations has been the lack of analysis of the results and the failure to attempt to find the causes of variations or to associate them with other physical properties. Unless and until this is done there will always be a tendency to make further tests, and differences in overall results are now often explained as due to errors or inaccuracies in the other man's methods or work. Attempts to derive ways for final analysis are of value for this reason rather than that they give more accurate values for field use, since the variations in commercial materials have usually a greater possible range than have the errors introduced by approximate analysis of results. A test such as that for Fig. 5 would take at least four weeks, and a great deal of care, and it should therefore be worth while to work up the results very thoroughly, and it should at least be expected that the investigator should have a clear idea of the fundamentals connected with his work.

Replying to Mr. McMillan, the labor in working up the curves is not great and a tabular form of computation can be used which makes it easy. For the tangents it is only necessary to mark off points for each on the lower and right hand axes, and as the tangent is the ratio of these lengths (taken of course in their scale values), the straight edge need only be set parallel to the true tangent. Having the test curves, the tabulated arithmetical work for the others, as well as the check back for their corrections does not cover a sheet of paper.

The test curve M in Fig. 5 is for the B.t.u. per sq. ft. of inner surface of sample, per hour, and the T_2 is the actual inner surface temperature, and not that of the pipe. The value of S to use with this is 1.77.

I do not see why the N curve should be expected to be any definite shape, nor why it cannot be a straight line. The test points of Fig. 5 are shown, and it will be noted that the T_2 scale used is five times that of T_1. All the test results I have worked up by this method have given straight lines, and it is interesting that most of those for 85 per cent magnesia, necessitate the curvature at the end in order to reach the air temperature point, whereas that for felted asbestos ran through it. As the k values with the thickness used are small, the lack of knowledge of the exact end shape has not caused any trouble.

As stated in the paper Fig. 8 could not be printed to a scale to make it a refined, workable diagram, and consequently only one scale for the surface loss law is given. There could, however, be several such scales drawn, and similar temperatures connected by curves so that one could transfer by eye the scale it was desired to use.

It is of course very desirable that the test result should give smooth curves, which in the illustration used means that the air temperature should be kept constant during the tests. If it varied to any extent it would cause bumps in the test curves which would make the work of analyzing them more difficult.

Mr. Heilman's surface loss equation is empirical and in his paper he limits its use to 20 in. diameter. Mr. McMillan is therefore not correct in applying it to a flat surface.

No. 640

AIR CONDITIONING FOR SAUSAGE MANUFACTURING PLANTS

By M. G. Harbula, Chicago, Ill.
Member

THE sausage as an article of food, date from remote antiquity. It is said to have been mentioned in history in connection with the ancient Romans as far back as the year 80 B. C. Yet, up to recent times the methods of making it were somewhat haphazard.

It was commonly believed until a few years ago that sausage-making was the last resort of beef and pork sellers when they were unable to dispose of certain cuts and trimmings of meat. Being unsalable as pieces the idea was common that they were carefully chopped into indistinguishable fragments highly seasoned and carefully concealed inside the *bags of mystery,* as sausage casings were then sometimes called. All of this, however, has since changed. From the inception of the modern packing house about 40 years ago, the business of sausage-making on a commercial scale has developed into one of large proportions. Packers who regard the sausage department as one that can always show a profit are the most successful ones; furthermore, many substantial businesses have been built around this department alone.

Sausage is made in a great many varieties and the formulae for the manufacture and preservation of the various kinds are multitudinous. For the purposes of this paper, sausage will be classified under two general headings, namely summer or dry sausage which is a product that is smoked and dried, but not cooked, and domestic or fresh sausage which includes all other varieties.

Air conditioning equipment for the maintenance of proper atmospheric conditions in rooms devoted to the various steps in sausage manufacture, from the handling of raw meat trimmings and meat cuts to the finished product, is playing a very important part in this rapidly expanding industry. In some of the phases of manufacture, conditioned air, properly distributed is absolutely essential at all times and seasons in order to turn out a product of uniformity and quality. In each and every building and room devoted to sausage making, conditioned air has been found a profitable investment.

Paper presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Buffalo-Detroit, June, 1922.

DOMESTIC OR FRESH SAUSAGE

This sausage is made mostly from beef and pork trimmings and various parts of the animal organisms. The usual process of manufacture is as follows:

The meats or trimmings used are comminuted in grinding machines; some manufacturers chop instead of grind the meats used in certain of their brands. After grinding, the spices are thoroughly worked into the mass of meat in a mixer. The meat is then transferred to what is usually termed the green-meat cooler where it remains for a period of

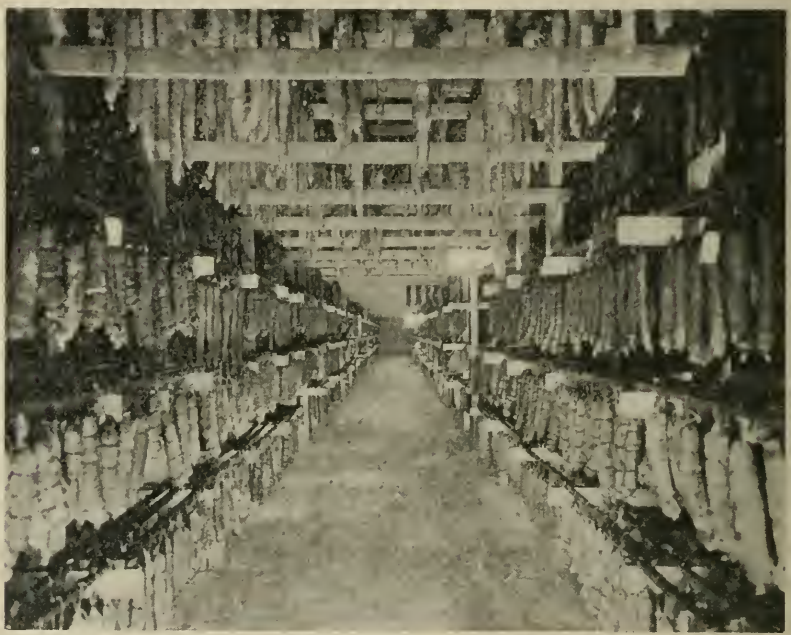

FIG. 1. A LARGE SAUSAGE DRYING ROOM

time, varying from a day to a day and a half, during which a curing is effected. The meat is then ready for stuffing into regular casings, cloth bags, or other containers as required for the trade. After the sausage is stuffed it is smoked, then cooked, after which it is rinsed with hot water, then chilled with cold water, dried off and hung in the dry-hanging room until ready for shipping.

SUMMER OR DRY SAUSAGE

It is generally admitted by packers and sausage makers that of all diversified products turned out of their establishments, summer sausage has been the most difficult to produce. Being strictly air-dried and requiring a long time for production, variable atmospheric conditions affect it adversely. The successful producers are those who have special facili-

ties for maintaining uniform air conditions in the various rooms devoted to the manufacture of this product.

Before the days of the existing pure food laws some varieties of summer sausage could be made during the greater part of the year, due to the fact that questionable preservatives were used quite liberally. The chief agent which the manufacturer is permitted to employ today is saltpetre and some preservative help obtained from the spices. Summer sausage could always be manufactured after a fashion during the winter months of the year; in fact, it was always considered in the earlier days

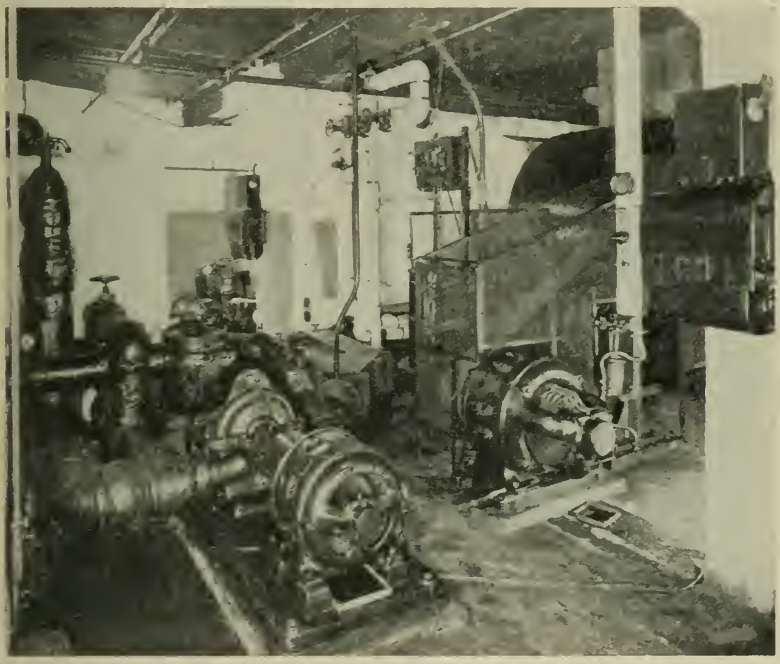

FIG. 2. MACHINERY ROOM OF AN AIR CONDITIONING SYSTEM

of the business that the only period during which a high grade product could be made was in cold weather. During the summer months production was practically suspended unless the manufacturer had facilities to make him independent of outside weather conditions.

Summer sausage is made from selected meats and trimmings. For some varieties of this product the meats are coarsely ground, then chopped on a rocking cutter together with the particular seasoning selected. The materials for coarse sausage are then worked in a mixer for several minutes. This part of the work is carried on in the manufacturing room. The meat mass is then transferred to the green-meat cooler, where it remains 12 to 48 hours while the spices, salt-petre and common salt permeate the product, after which it is returned to the manufacturing room and stuffed in casings selected for the particular variety. After stuffing,

the product goes into the green sausage hanging room for 1 to 3 days while the surface moisture is dried off. It is then ready for either the smoke houses or the drying room, the destination depending on the variety. Certain types require smoking in various degrees and temperatures prior to transfer to the drying room; others are unsmoked and pass directly to the drying room where the final cure is effected, the time varying from a few days to 2 and 3 months and more. When the sausage is properly cured it is washed and dried if necessary and moved to the holding room, which is essentially a storage cooler. When orders are to

FIG. 3. FAN WITH SPRAY TYPE DEHUMIDIFIER FOR AUTOMATIC TEMPERATURE AND HUMIDITY CONTROL

be filled the finished product is taken to the packing and shipping room, and boxed for shipment.

In some establishments every step of summer sausage manufacturing is carried on in rooms that are maintained at a comparatively low temperature the year round, and this is as it should be. Conditioned air controllable in respect to its volume, temperature and relative humidity is practically indispensable and many plants today are using it.

The green-meat cooler should be kept clean, dry and airy and at a comparatively low temperature, 34 to 38 deg. fahr. being best; the relative humidity should be kept high as possible. With the older methods of cooling, dripping ceilings and excessive dampness are common; the air not being in active circulation results in the creation of objectionable odors. Conditioned air is not only desirable, but by many is considered

necessary; both temperature and relative humidity can be maintained uniformly. In the absence of sunlight the constant circulation of air keeps the cooler dry and assists in holding conditions up to the rigid standard of sanitation which must be adhered to throughout the entire process of sausage-making.

The manufacturing room should be kept at the lowest temperature consistent with good working conditions. The temperatures most satisfactory are a compromise, 50 to 55 deg. fahr. being found best. From the standpoint of the product much lower temperatures would be desirable, but their influence on comfort and efficiency of the workers is adverse and production is reduced. With conditioning apparatus pure air is kept in circulation constantly; heat from various sources, such as

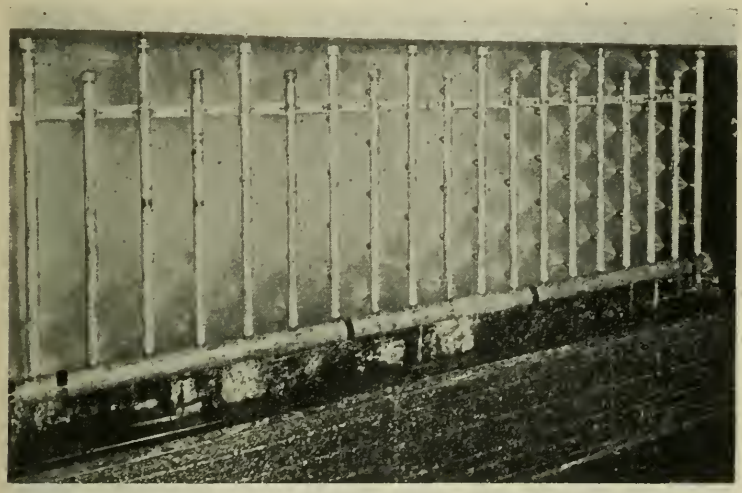

FIG. 4. SPRAY TYPE DEHUMIDIFIER IN ACTION—AIR INLET END

the workers, lights, motors and stuffing machinery, is taken up by the air as it is generated. Keeping the meats in a sweet condition is facilitated and the conditioned air also has an energizing effect on the employes. There is a large amount of handwork in connection with the stuffing of sausage and heavy demands are made upon the speed and dexterity of the workers.

Considerable care must be exercised in the design of the air distributing system to prevent drafts on the workers; the air volume being large and the entering temperature comparatively low.

In the green-sausage-hanging room control of temperature and relative humidity is important and equally so is a positive circulation of air. The temperatures maintained in various establishments are 54 to 60 deg. fahr. with relative humidities ranging from 55 to 65 per cent depending on the kind of product and the individual ideas of the sausage maker.

It is here that the sausage takes on its initial color. Color being a major selling point, it is important that all factors governing its develop-

ment be considered in the minutest detail. While in this room the product is carefully watched as it must not become too dry, nor be removed while in a slimy condition. Slime is more detrimental than excessive dryness since slimy sausage will not take smoke and may sour. On the other hand, if it is allowed to dry too rapidly or too much, case-hardening results and smoke will not penetrate the casing.

The usual methods and means for controlling air conditions within this room are more or less crude and unreliable and the quality of the sausage, if turned out at all, is dependent upon the care of the employee in charge of the process. With conditioned air properly circulated, dis-

FIG. 5. SUPPLY AIR DUCT WITH OUTLET BRANCH

tributed and maintained at uniform temperature and humidity, the processing that the sausage undergoes in this room is constant, uniform and certain during all seasons of the year; furthermore the manufacturer is less dependent on the human element for results, a factor not to be lost sight of.

The importance of proper circulation and distribution of air should be emphasized. The distributing system should be carefully designed so as to prevent drafts on the product and local stagnation. The airiness demanded makes this quite a problem.

As mentioned heretofore, some varieties of summer sausage are smoked, while others are not. Smoking, in general, is a combined drying, curing and preserving process. In smoking meats in any form, whether sausage, hams, breakfast bacon, dried beef or other meats, the objects aimed at are, first, to smoke the products as quickly as possible, consistent with

good work; second, to keep shrinkage as low as possible; third, to turn out a product which is dry, handles firmly and has a pleasing color.

Successful smoking is dependent, to a considerable extent, on proper weather conditions. If the weather is humid or damp, a longer time is naturally required and if the temperature within the smoke houses is raised to dry the air, that is, to lower the relative humidity, overheating results, which is responsible to a great extent for dull color. Smoking in a temperature higher than that best for a given product is generally unsatisfactory and the texture of the product, as a result, is never firm, as it should be.

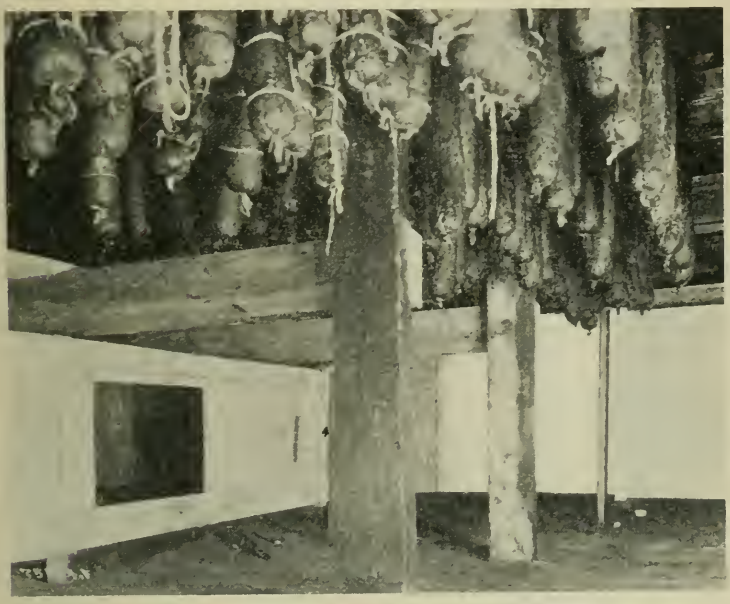

FIG. 6. RETURN AIR DUCT WITH INLET OPENING

A common source of annoyance to the smoke house attendant is the condensation of moisture on the product, whereupon the smoke fails to take effect and the product has a striped and discolored appearance.

Since weather conditions have a great deal to do with the smoking of sausage and other meat products, smoke houses should be properly arranged and provided with equipment for controlling temperature and humidity. The smoking temperature of practically every kind of sausage varies and the equipment should be capable of a reasonable range of variation. With proper conditioning apparatus the time required for smoking may be lessened and a constant output of a uniform product anticipated.

After smoking, or, in some cases, immediately after treatment in the green-sausage-hanging room, the product is transferred to the drying room, where it receives its final *cure* and is reduced to the stage of dryness required for the different varieties.

Successful drying is dependent on a most favorable atmospheric con-

dition. Air at proper temperature and relative humidity is absolutely necessary. The sausage in drying must dry from the inside, outwardly. The surface of the casing must be kept in such a condition that moisture from the interior of it may be extracted gradually and constantly. Sausage correctly made and stuffed will contain no hollow spots when properly dried and the meat and other ingredients will have set into a firm, compact mass. If the air within the room is too dry, case-hardening results and only the meat nearest the casing will become dry, leaving the interior in a soft, moist condition. With the moisture thus confined interior mold may develop or the sausage may sour and spoil; oftentimes an entire run of sausage is relegated to the rendering tank. Interior mold is particularly damaging since its presence may not be discovered until in the hands of the consumer.

Should the air be too humid excessive surface mold develops. Some manufacturers do not object to a light surface mold; in fact, certain varieties of product require it. Surface mold is removed by washing in a special machine, but the cost is a considerable item, especially if the mold growth is heavy. This factor, as much as any other, has influenced manufacturers to adopt methods and means to reduce mold growth to a minimum, or to eliminate it entirely.

Attempts have been made to continue production throughout the warmer months of the year, but after suffering loss of product in some cases, loss of market due to poor product in others, and being unable to maintain a constant output, manufacturers realized the necessity for equipment that would maintain constantly in their drying rooms an air condition, predetermined as best.

The temperatures maintained in sausage drying rooms range from 48 to 56 deg. fahr. and the relative humidities from 65 to 80 per cent, depending upon the character of the product.

The air conditioning equipment that has proved satisfactory for this work consists essentially of a fan or blower, with means of driving same at varying speeds; an efficient spray type dehumidifier for treating the air; apparatus for cooling the medium sprayed; a pumping unit for circulating this medium; heating surface for warming the air when necessary; temperature and humidity control apparatus with safety devices for the prevention of over-humidification or overheating. Means for recirculating the air and for the use of outdoor air to save refrigeration in winter, together with suitable air relief facilities, should be provided. Lastly, but very important, is a properly designed system for distributing the conditioned air within the drying room and a collecting system for returning the air to the machinery room to be reconditioned. All these elements are interdependent and constitute in their entirety a system.

The system must be flexible and must lend itself readily to control. The only satisfactory method for maintaining temperature and humidity conditions within the close limits demanded in this work is in connection with the use of a spray type dehumidifier. This apparatus is the heart of the system.

Aggregations of equipment differing from the combination before mentioned have been used in the attempt to maintain conditions desired in the drying room, but with indifferent success.

In many of the applications of air conditioning it has been said, and, properly so, that an air conditioning system is no better than its distributing system which truly constitutes the arteries and as such they must accomplish their purpose. Advantages gained by conditioning are sometimes nullified by improperly designed and installed distributing systems. In no work is this more true than in the application to sausage processing. The design for the drying room system particularly must be correct in every detail. The method of introduction of air and its removal are important. The introduction should not cause drafts on the sausage and removal must be done in a manner which will prevent short-circuiting.

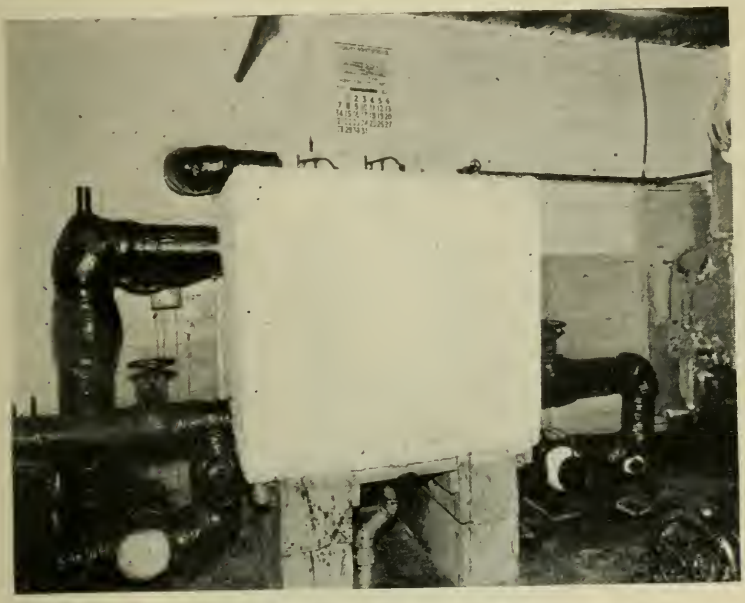

Fig. 7. RESERVOIR FOR MEDIUM SPRAYED IN DEHUMIDIFIER AND AUTOMATIC VALVES FOR CONTROLLING THE SUPPLY OF MEDIUM

In other words, a definite quantity of air must be distributed uniformly so as to pass over every piece of sausage in the room.

Each drying room is an individual problem for the air conditioning engineer. These rooms necessarily vary in each establishment in location, size, proportions, construction, arrangement, etc., consequently, the details of design and arrangement of air conditioning equipment vary in every case.

Air conditioning equipment is indispensable for drying rooms and is a profitable investment. For instance, the experience might be related of a concern that was contemplating the equipping of their sausage dry rooms with air conditioning equipment. One such room was equipped and a record of costs of the entire installation was kept, which were eventually checked against the results obtained. The results demonstrated that the equipment paid for itself in the first four months of its operation. This

was due to many causes. It was found unnecessary to change the sausage from one location to another to have it dry properly, labor thus being saved; sausage molded less rapidly and where proper precautions were observed mold was practically eliminated, thus reducing expense incidental to washing; and it might here be said that washed sausage never has the bright attractive color and appearance of unwashed, mold-free sausage—a factor of great weight because the American public which is rapidly developing a taste for this product buys first of all on appearance and continues to buy if the quality is there in addition. With some sausage the smoking and also the drying and curing time was reduced, this also effecting a considerable saving; there was no loss of product, which before could always be expected; the color was better and the quality was so improved that the sausage from this one room had a greater market value, and it was also found that the sausage *held* better. The product was uniform and output was increased without hesitation through fear of unfavorable weather conditions, and shipping dates could always be anticipated. These and other deciding factors resulted in this concern similarly equipping their other sausage drying rooms.

Air conditioning has been successfully applied to drying rooms of both large and small capacity. The author has designed and installed air conditioning equipment in dry rooms ranging in size from those with hanging space for 50,000 lb. upward to more than 1,000,000 lb. of sausage, with gratifying results.

The holding room should be carried in the neighborhood of 40 deg. fahr., and should be kept dry. The packing and shipping room is usually kept at a somewhat higher temperature, mainly because of the presence of workers; 56 to 62 deg. fahr. being satisfactory. The air should be kept sufficiently dry to prevent condensation of moisture on the cold product coming from the holding room.

The schedule of operations mentioned in connection with the production of sausage is that more generally followed by successful manufacturers. At a time when far-seeing executives of meat-packing establishments saw and realized the possibilities of the sausage-making business, the parts of their plants that were devoted to this business were made over so that they could produce a greater variety on a larger scale and in any weather. In some plants it was impossible, for various reasons, to have separate rooms for each operation. Within the past few years many establishments have reconstructed their sausage departments, in some cases remodeling old buildings, in others adding new space and new buildings so as to confine each phase of the business to a separate enclosure. In the development to this stage air conditioning has been a dominant factor, being applied first to dry rooms. Its application has been extended to practically every room connected with the manufacture of summer sausage as well as to those devoted to the making of domestic or fresh sausage. Today when a manufacturer is contemplating the modernization of his sausage department so as to permit him to successfully compete with a product of prime quality that can be turned out on a quantity basis at all times, air conditioning equipment, having proved itself and having been found indispensable, is made part of the construction plans and is designed and provided for accordingly.

In Memoriam

Names	*Joined the Society*	*Died*
John F. Kennedy	September, 1921	February 14, 1922
Alfred J. Wells	June, 1917	April 17, 1922
Clarence A. Nielsen	February, 1921	August 30, 1922
Harrie D. Crane	Charter Member	September 15, 1922

Harrie D. Crane

HARRIE D. CRANE

Harrie D. Crane, a prominent heating and ventilating engineer of Cincinnati, O., passed away at his home in Avondale, September 15, at the age of 68, after an illness of several years. He was president of the M. H. Crane Estate, founded by his father and was identified with some of the largest heating and ventilating work in that section of the country.

Mr. Crane was born in Cincinnati, September 26, 1854, and his preliminary education was received at Chickering Institute. He always took an active interest in association affairs and served two terms as president

of the Heating and Piping Contractors' National Association in 1894 and 1895. As a charter member of this Society, and its President in 1903, he was intimately known and universally liked by the older members, who knew his high ideals and assisted in his efforts for the uplift of the profession.

In his own community, Mr. Crane was active in civic affairs being a prominent member of the Cincinnati Business Men's Club, serving on its board of directors and numerous committees. In his earlier years, he was active in athletics and retained a keen interst in sports throughout his life. He was a member of several fraternal organizations, was a Shriner and belonged to the Hanselmann Commandery.

His widow survives and will continue the business of the M. H. Crane Estate. Burial was in Spring Grove Cemetery, Cincinnati, September 18.

The passing of Mr. Crane will be keenly felt by members of the Society as well as his many intimate friends and associates. A tribute to the high esteem in which he was held was the fact that every man in the heating and ventilating profession of Cincinnati was represented at his funeral.

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